COMBUSTION ENGINES

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COMBUSTION ENGINES

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PREFACE

The term *combustion engines* is taken to include the commercially important types of heat engine in which the working fluid consists of the products of combustion of hydrocarbons and air rather than steam or other vapor. Although this book is primarily concerned with reciprocating engines, the problems of combustion, practical thermodynamic analysis, fuel metering and injection, lubrication, etc., are so similar in gas turbines that it has proved to be quite easy to cover almost simultaneously most of the thermal and mechanical problems of gas turbines. The aerodynamics of gas turbines is a subject in itself and could not properly be included.

The text has been designed primarily for the use of engineering college seniors and young engineers in the industry. To satisfy the needs of this group it seemed best to develop the idealized relationships from elementary theory and then show how these are related to methods that have proved very satisfactory for solving practical engineering problems. An exhaustive treatment of all the detailed investigations carried out in the combustion-engine field has not been attempted. In particular, much that is largely of academic interest has been omitted in order to avoid obscuring relations of great practical importance.

A familiarity with college chemistry, physics, thermodynamics, and fluid mechanics is essential to a full understanding of the text. A similar knowledge of metallurgy and machine design has also been assumed but is not of critical importance because the detailed problems and methods of engine-part design are not discussed. The latter are subjects in themselves and are usually treated in machine-design and stress-analysis courses.

The author is grateful to the many firms that helped provide illustrations and is especially indebted to the National Advisory Committee for Aeronautics (NACA) for its great contribution in this regard. As indicated by the many bibliographical references, the book could never have been written without the wealth of material available in the literature. An equal debt is owed to friends and

colleagues who have contributed generously of their time and experience by offering many suggestions as to the scope and content of the book as well as by making a tremendous number of detail corrections and additions to the manuscript. Professor H. J. Buttner, together with C. D. Miller and C. Dickerson were exceptionally helpful. C. A. Bierlein, R. V. Brown, F. A. Dietz, R. N. DuBois, R. C. Juvinal, R. L. Larson, J. Meinzinger, E. S. Olmstead, R. Quaint, W. M. S. Richards, J. L. Ross, V. E. Schafer, Jr., W. B. Seaver, T. J. Walsh, and A. L. Wittmer all gave invaluable assistance.

ARTHUR P. FRAAS

RIO DE JANEIRO, BRAZIL
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CHAPTER I

ENGINE TYPES AND CONSTRUCTION

Power has become both an outstanding characteristic and one of the principal bases for life in twentieth-century America. Power—to drive passenger cars, bulldozers, trucks, tanks, and airplanes, to furnish electrical energy for lights and radios, in short, to keep the whole complex machinery of the age running fast and smoothly—may be derived from any one of a number of prime movers. The relative practical importance of each is indicated by Table 1, which shows the nature of the power resources of the United States at the time of writing. In terms of the total power-producing capacity of the country, Table 1 shows that the internal-combustion engine is by far the most important of the various prime movers, for it constitutes over 90 per cent of the nation's installed horsepower. Internal-combustion engines supply all the installed horsepower for the automotive and aircraft groups, most of that for agricultural equipment, and a fairly large part of that under the other groups except the three covering electric power plants. Only about 4 million horsepower was obtained from oil and gas engines in the latter groups.

There are a number of reasons for the dominant position of internal-combustion engines as demonstrated by Table 1. Probably the most important is the use of air as a working fluid in the thermodynamic cycle on which the engine operates. This has made it possible for internal-combustion engines to take in a fresh charge of air for the cycle, add heat to it by burning fuel in that air, and then—after completion of the cycle—throw the working fluid away and start with a fresh charge, thus eliminating bulky and expensive heat exchangers. Further, the ready availability of air has made a reservoir of working fluid unnecessary, thus effecting another major saving in bulk, weight, and cost. The net effect has been to reduce the weight, bulk, and initial cost of internal-combustion engines to values much below those for any other type of prime mover, except in a few types of application.

The relative distribution of the internal-combustion engine to the various applications may also be seen from Table 1. Passenger cars,

TABLE 1. POWER CAPACITY OF PRIME MOVERS IN THE UNITED STATES

	bhp*
Electrical central stations	50,000,000
Industrial power plants	25,000,000
Electric-railway plants	2,500,000
Agricultural equipment	80,000,000
Passenger car, truck, and bus	2,400,000,000
Locomotives	100,000,000
Civil aircraft	4,000,000
Military aircraft	60,000,000
Naval vessels	70,000,000
Merchant marine	10,000,000
Total	2,801,500,000
Summary of Power Capacity by Type of Prime Mover	
Noninternal-combustion engines	250,000,000
Spark-ignition engines	2,540,000,000
Diesel engines†	12,000,000

* Brake horsepower.

† Approximately 50 per cent of diesel-engine capacity is stationary or semimobile, 25 per cent is marine, and 25 per cent is used in trucks, tractors, or locomotives.

trucks, busses, and agricultural equipment account for the bulk of the internal-combustion-engine installations. All these are vehicles that have often been collectively classified as *automotive* or *self-propelled.* The problems associated with the manufacture, operation, and maintenance of the engines in these vehicles are very similar. All require a relatively lightweight, ruggedly dependable power plant that will operate well under any conditions of climate or weather with little protection. The engines must be flexible to meet the operating requirements of the moment. That is, not only must an automotive engine run well at any point in a wide range of speeds, but also it must operate equally well at any speed under whatever load the occasion may demand. A passenger-car engine, for example, may be called on to deliver a high power output when going uphill; yet a few minutes later at the same speed it may have to operate as a brake to absorb power as the car goes downhill. Easy starting in any weather, low fuel consumption, and trouble-free operation are other requirements that all automotive engines must meet. It is interesting to note that practically all the basic problems which arise in connection with the use of internal-combustion engines in other than automotive applications have their counterpart in the automotive field, although the reverse is not always the case. Partly because of this, and even more because of the relative size of the industry, this book will be concerned more with automotive engines than with any other type.

TYPES OF APPLICATION

One way in which internal-combustion engines may be classified is on the basis of the various uses to which they are put. While there are no absolute standards for classification, the following groups give one means for differentiation:

Passenger Car. Of the variety of motor vehicles in widespread use on our highways, approximately 85 per cent are passenger cars. Many of the balance including station wagons, light trucks, and similar vehicles are powered with engines from passenger-car production lines. Engines for these applications usually develop 50 to 120 hp.

Truck and Bus. Large trucks and busses require a power plant capable of operation under heavy loads for long periods of time. Although production models of passenger-car engines are commonly used in the smaller trucks and busses, the larger vehicles require heavy-duty engines designed expressly for that purpose. Most of these engines develop 50 to 250 hp.

Agricultural and Earth-moving Equipment. Tractors, bulldozers, power shovels, and road-construction equipment often make use of truck engines. However, the very heavy loads and high powers required by the larger units have brought about the development of engines especially suited to this type of work, with the result that the years since about 1930 have seen advances in earth handling which are as spectacular in their way as the revolutionary strides made in the development of large aircraft.

Portable and Light Marine Power Plants of 30 to 250 Hp. The large volume of production of engines for the passenger-car, the truck and bus, and the agricultural-implement fields has made the cost of these engines much lower than that of any other type of power plant. As a result, they have been widely used as portable and stationary power plants to drive air compressors, generate electricity for light, power, and arc welding, and supply power for construction equipment such as concrete mixers and for other similar applications. Most of the light marine engines for powerboats appear so much like truck and bus engines that they are almost indistinguishable from them, except for the substitution of a reverse gear for the transmission and the elimination of the fan.

Railroad, Marine, and Stationary Power Plants. The primary requirements of exceptionally high dependability and low fuel-consumption rate with secondary requirements of moderate engine weight and volume and a moderate degree of operational flexibility charac-

terize a number of applications of which railroad-locomotive, ship-propulsion, and stationary power plants are by far the most important. While the same identical engine model is seldom used in more than one of these three types of application, the differences between the equipment used in the three fields are quite small. The engine sizes in most instances range from 200 to 2,000 hp. Although much larger units are available, their number is relatively small.

Light Aircraft. Piston-type aircraft engines may be roughly divided into two groups, *unsupercharged* and *supercharged* engines. The problems of the unsupercharged engines widely used in "light planes," except for weight considerations, are not greatly different from those of motor-vehicle engines except that they are almost universally air-cooled instead of water-cooled. Engines very similar to but smaller than these light-plane engines have been used in motor vehicles, particularly in Europe.

Heavy Aircraft. While large highly supercharged piston-type aircraft engines present many detail problems too specialized to be considered in this book, the basic operating principles are the same as those of motor-vehicle engines. Turbine, turbojet, etc., types of aircraft engine are also coming into use and are making the aircraft field a proving ground for the gas turbine.

Miscellaneous. Motorcycle, outboard-motorboat, lawn-mower, small-agricultural-implement, and other specialized applications require small engines that present problems similar to those of passenger-car and/or unsupercharged aircraft engines. They form a small but interesting segment of the automotive field.[1*]

ENGINE DESIGNATIONS

Engines are classified in a number of ways in addition to that of the type of application.

Reciprocating and Turbine Types. Practically all internal-combustion engines have been of the *reciprocating* type, in which the hot gases of combustion are allowed to expand in a cylinder and do work on a piston. The *gas turbine*, however, is coming to be used to an increasing extent in applications requiring units of 1,000 hp or more, *i.e.*, large aircraft, locomotive, marine, and stationary power plants.

Many of the problems associated with reciprocating and turbine types of power plant are closely related. Those of combustion, fuels, fuel metering, lubrication, air compression and supercharging, installation, and the development and fabrication of parts capable of operation in high-temperature gas streams are examples. While this text is primarily concerned with the piston type of engine, approximately 40 per cent of the material is also applicable to gas-turbine work. The more important problems peculiar to gas turbines are discussed briefly in a special chapter designed primarily to present performance characteristics of gas turbines relative to those of piston engines and hence the consequent advantages of each.

* Superior numbers refer to bibliographical references at the end of each chapter.

Two- and Four-stroke Cycles. Internal-combustion engines of the reciprocating type may operate on either a two- or a four-stroke

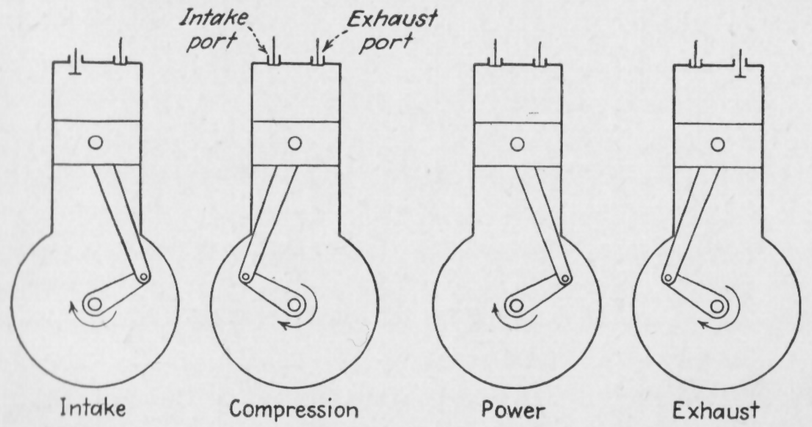

FIG. 1. Phases of the four-stroke cycle.

cycle. Figure 1 shows the events in the four-stroke, or Otto, cycle. In the ideal case the intake valve opens when the piston reaches top dead center. It remains open to allow the fresh charge to flow into the cylinder as the piston moves downward on the *intake stroke* to bottom dead center. It then closes so that the charge is compressed on the *compression stroke*. Ignition and combustion of the charge near top center bring about a large increase in pressure to force the piston downward on the *power stroke*. The exhaust valve opens near lower dead center to relieve the pressure remaining in the cylinder. It remains open until the subsequent upward *exhaust stroke* of the piston has scavenged the cylinder by mechanically forcing out most of the remaining exhaust gas.

Figure 2 shows the events in the *two-stroke cycle*. The piston compresses the charge in the cylinder, ignition occurs as the piston nears the top of its stroke, and the piston is driven downward by the high pressure of the heated gases. As it approaches the bottom of its stroke in the type shown, the exhaust valve opens to permit the

exhaust blowdown to occur, *i.e.*, to allow enough of the gas in the cylinder to escape to drop the cylinder pressure to atmospheric. By the time the exhaust blowdown has been completed, the piston uncovers the intake ports to permit air under pressure to rush into the cylinder,

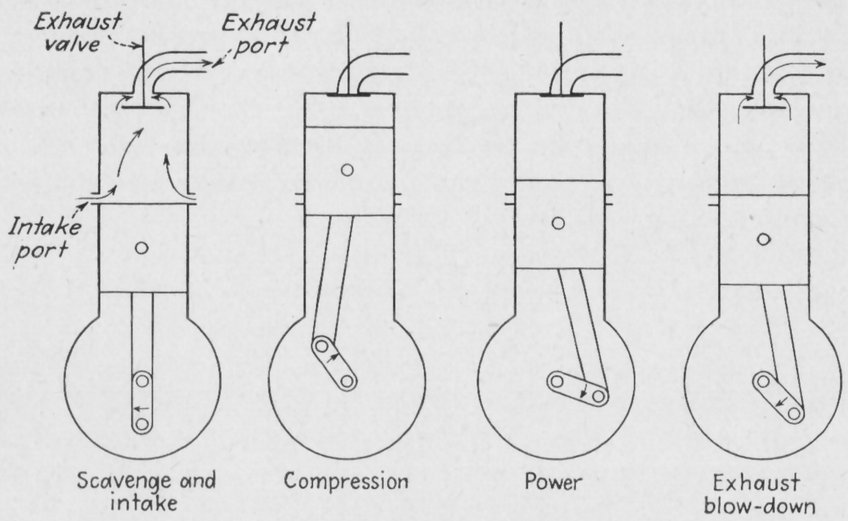

FIG. 2. Phases of the two-stroke cycle.

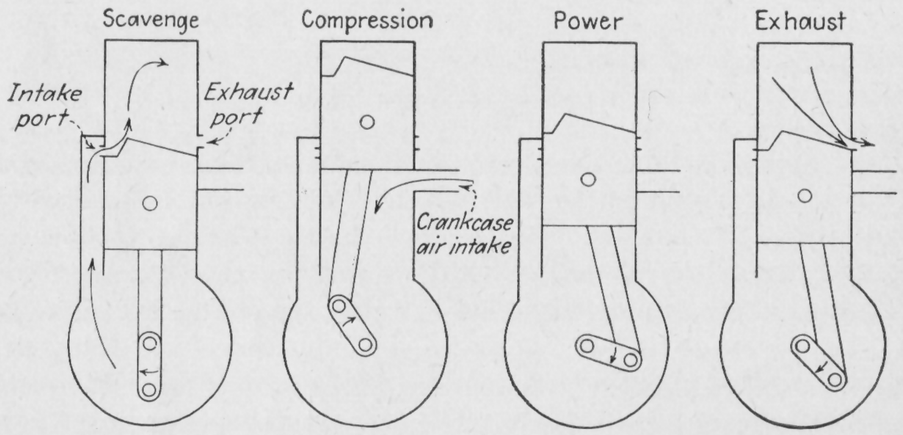

FIG. 3. Loop-scavenged piston-ported cylinder. Arrows show direction of air flow during the exhaust blowdown and scavenge phases of the cycle.

push out most of the remaining exhaust gas, thus *scavenging* the cylinder, and fill it with a fresh charge before the piston has moved far enough upward on the next compression stroke to close off the intake ports. Other port arrangements may also be employed to give the same sequence of events.[2] Figure 3 shows the arrangement commonly used in both outboard-motorboat and model-airplane engines. The exhaust port is placed higher in the cylinder than the intake port so

that it is uncovered first to permit the exhaust blowdown to occur. When the intake port is uncovered, it allows the fresh charge that has been compressed in the crankcase by the downward motion of the piston to rush into the cylinder. A deflector on the top of the piston directs the flow upward along a loop-shaped path to give better scavenging of the exhaust gas remaining in the cylinder after the blowdown. Figure 4 shows an opposed-piston engine in which the crankshafts are timed a little out of phase so that one piston first uncovers the exhaust port, permitting the exhaust blowdown to occur before the other piston uncovers the intake port. A number of models of diesel engine use this arrangement.

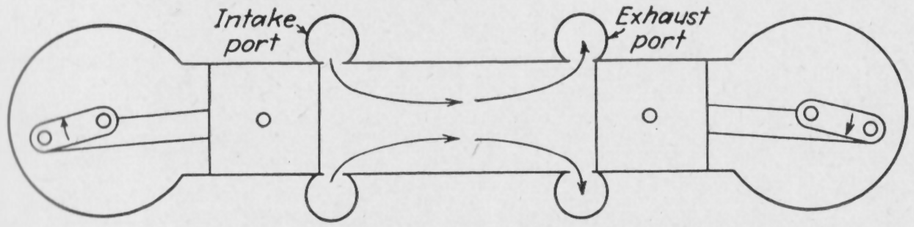

FIG. 4. Opposed-piston piston-ported cylinder.

Types of Valve Mechanism. Figures 1 to 4 show two methods of valving the working fluid, *i.e.*, *poppet valves* and *piston porting*. Other types of mechanism have been employed, but only one type, the sleeve valve, has been used in any production engines. Several varieties of sleeve valve have been built successfully.[3,4] All depend on a sleeve reciprocating between the piston and cylinder to control the intake and exhaust events.

Method of Ignition. Those internal-combustion engines which depend upon an electric spark for ignition of the charge are known as *spark-ignition* engines. The first successful engine of this type was built in 1876 by Otto, for whom the four-stroke-cycle spark-ignition engine is sometimes called the *Otto-cycle* engine. On the other hand, compression-ignition engines are those in which the high temperature obtained by the compression of air is depended upon to ignite the fuel when it is injected into the cylinder near the end of the compression stroke. Such engines are also called *diesel* engines after their inventor, Rudolf Diesel, who began to develop them in 1892.[5] As a point of interest, the largest internal-combustion engine to have been built at the time of writing is a 22,500-hp diesel in Copenhagen, Denmark.

Cylinder Number and Arrangement. One of the most common ways of designating engines is by referring to the number and arrangement of the cylinders. While most of the early engines employed

only 1 or 2 cylinders, the majority of engines today have at least 4 and in some cases as many as 42 or more cylinders. A number of basic cylinder arrangements have been especially popular with designers. The arrangement most widely used in motor vehicles is that shown in

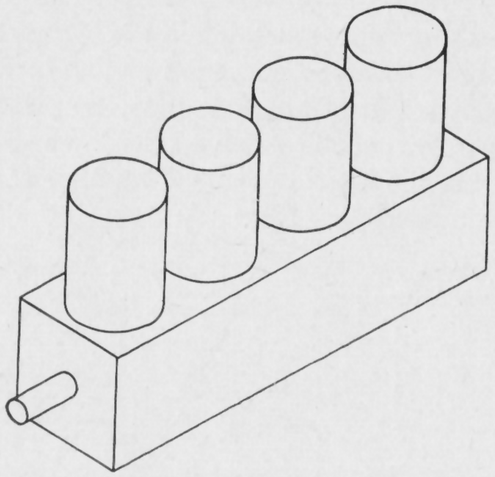

FIG. 5. Cylinder arrangement for a four-cylinder in-line engine.

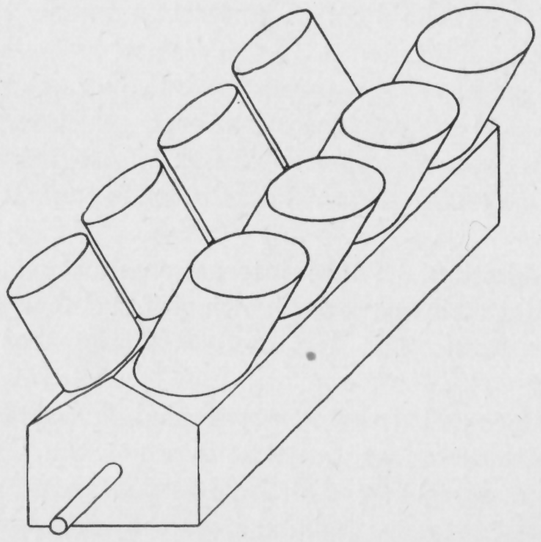

FIG. 6. Cylinder arrangement for an eight-cylinder V-engine.

Fig. 5, the *in-line engine*. A logical multiplication of this type is the *V-engine* shown in Fig. 6 in which two *banks* of cylinders are placed together on a common crankshaft. The majority of motor-vehicle power plants are comprised by 4-, 6-, and 8-cylinder in-line engines, together with 8- and 12-cylinder V-engines. Both the in-line engines and the V-engines are sometimes built to operate with the crankshaft

above instead of below the cylinders. When this is done, they are called *inverted* in-line engines, or *inverted* V-engines. Figure 7 shows a *horizontally opposed cylinder* engine of the type generally used in light planes. Note that the cylinders are not placed directly opposite one another but are offset so that alternate cylinders are lined up with alternate crank throws, there being one crank throw for each cylinder. A number of models of the *opposed-piston* engine have been built. These make use of the piston, cylinder, and crank arrangement indicated in Fig. 4. The *radial* engine has been widely used in aircraft and, in a few instances, in motor vehicles. Figure 8 shows a 5-cylinder

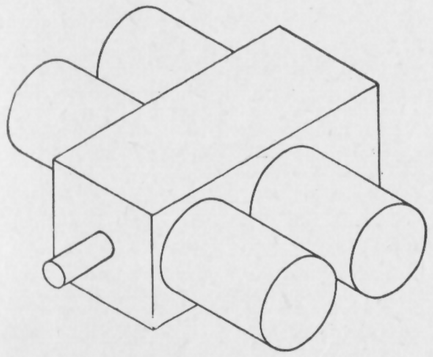

FIG. 7. Cylinder arrangement for a four-cylinder horizontally opposed cylinder engine.

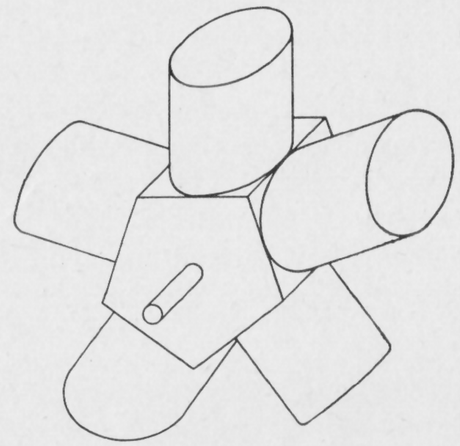

FIG. 8. Cylinder arrangement for a five-cylinder radial engine.

unit, although 7 and 9 cylinders are more often employed in a single row. In still larger radial engines, two or more rows of cylinders have been placed one behind the other to form the well-known 14-, 18-, 28-, etc. cylinder radial aircraft-engine types, which develop as much as 4,000 hp. Still other cylinder arrangements have formed the basis for many experimental and a few production engines, but their use has been quite limited.

Piston Displacement. The best indication of an engine's size is given by its *piston displacement*, or its *swept volume*, usually expressed in cubic inches. As the terms imply, they refer to the quantity of air displaced by the pistons in moving from their lower to their upper dead-center positions. The cylinder volume above the piston at top center is the *clearance volume*, while the difference between the clearance volume and the volume above the piston at bottom center is the *piston displacement* for one cylinder. The product of the number of

cylinders and the piston displacement of each gives the piston displacement for the engine as a whole. This may be expressed algebraically as

$$\text{Piston displacement} = \frac{\pi \times \text{bore}^2 \times \text{stroke} \times \text{no. cylinders}}{4}$$

Method of Cooling. An important distinguishing feature of an engine is the method by which it is cooled. Not only is there a considerable difference between the appearance of air-cooled and liquid-cooled engines, but the problems of design, development, and installation differ materially between the two types. The great majority of passenger-car, truck, and bus engines have been liquid-cooled because of the greater simplicity in design. Practically all motorcycle and light-plane engines have been air-cooled because of the weight savings made possible by the elimination of the coolant and radiator. Many advances in the technique of air-cooling have been made under the impetus of the Second World War so that the air-cooled engine may come to be used more widely in the automotive field.

ENGINE CONSTRUCTION

Certain basic features of construction have characterized practically all reciprocating combustion engines. While details differ with engine type and manufacturer, the many parts in any engine may be roughly grouped as follows: crankshaft, connecting rods, crankcase, pistons, cylinders, cylinder heads, valve mechanism, intake and exhaust manifolds, and accessory drives. In addition, various complex units including accessories, a fuel-metering system, a lubricating system, a cooling system, and—in spark-ignition engines—an ignition system are required. Since later chapters will cover the latter group of items, the following section is devoted primarily to the parts constituting the basic engine on which the other equipment is mounted. While 4- and 8-cylinder in-line engines and 12-cylinder V-engines are important types, the 6-cylinder in-line engine and the 8-cylinder V-engine are the most widely used. An example of each of these two is described in some detail in the following section as typifying automotive-engine construction, particularly that of passenger cars. The more important features of a diesel engine for heavy-duty truck and bus service, a light-plane engine, a radial aircraft engine, and a 16-cylinder V heavy marine diesel engine are also described briefly as being representative of their respective types.

PASSENGER-CAR ENGINES

The Six-cylinder L-head Automobile Engine. The general form and the arrangement of the various parts in one manufacturer's model of the most widely used type of internal-combustion engine—the 6-cylinder in-line L-head water-cooled spark-ignition engine—may be seen in Figs. 9 and 10. The nomenclature of the principal parts is

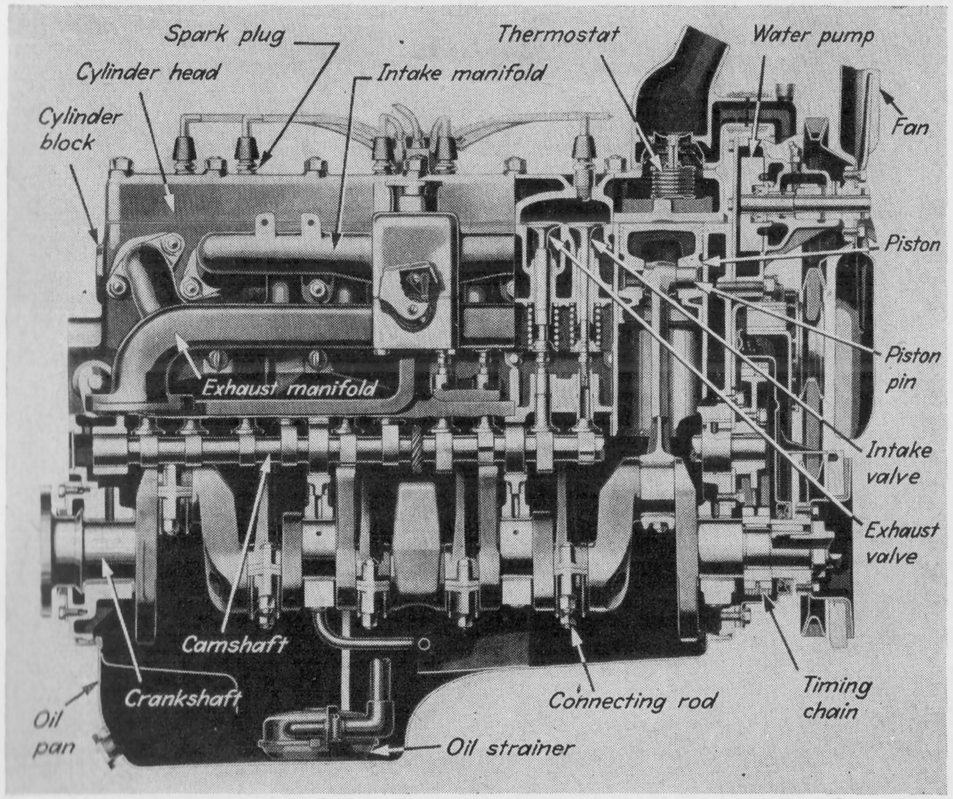

FIG. 9. Partial longitudinal section through a 1947 Dodge six-cylinder L-head passenger-car engine. (*Chrysler Corp.*)

also given. While details differ somewhat with the engine model and manufacturer, the particular engine described is representative.

In this engine, as in most engines, the translational motion of the pistons is converted into rotational motion by the connecting rods and crankshaft. While other types of mechanism have been employed,[6] the simplicity and reliability of the connecting rod and crankshaft have made its use almost universal. The details of the crankshaft for the engine in Fig. 9 can be seen in Fig. 11. This shaft has four *main-bearing journals*, which ride in four *main bearings* in the crankcase as shown in Fig. 9. The six *crankpins* each carry a *connecting* rod.

The rectangular sections between the crankpins and the main-bearing journals are called *crankcheeks.* These are often enlarged and extended to provide *counterweights* for balancing purposes. The journals are ground and hardened, but the surface of the cheeks is left unfinished.

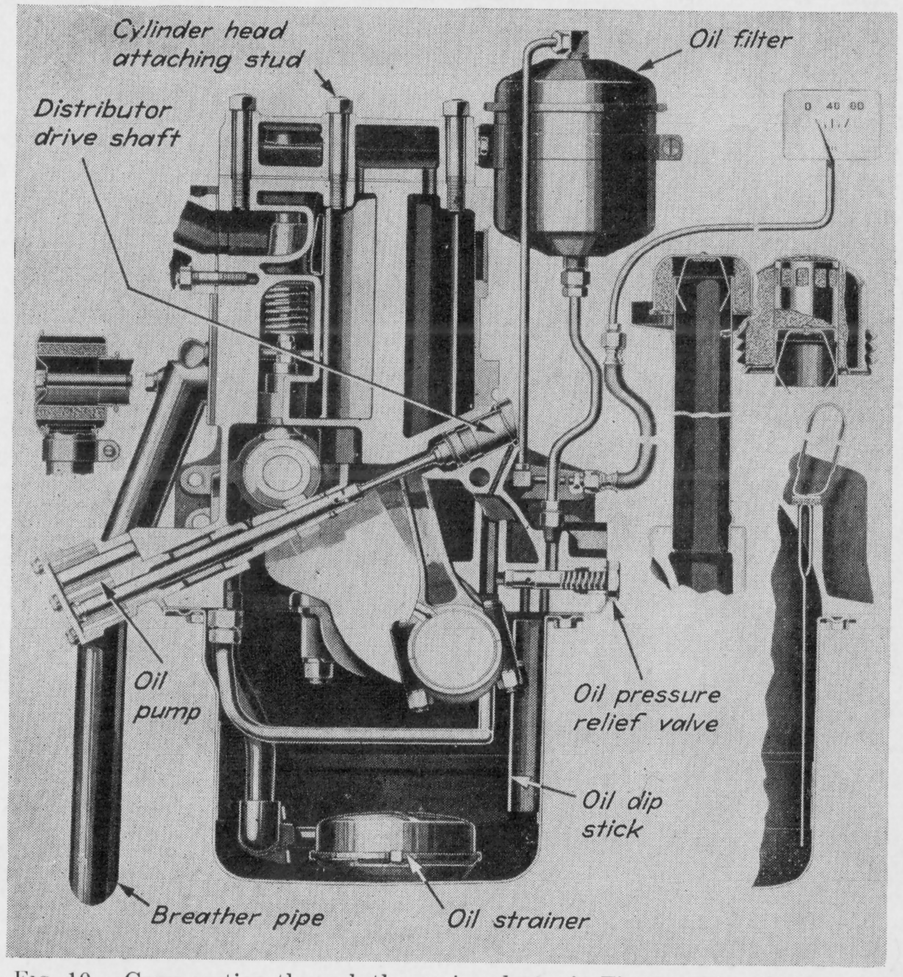

FIG. 10. Cross section through the engine shown in Fig. 9. (*Chrysler Corp.*)

The various parts that go to make up each connecting-rod assembly are shown in Fig. 12. *Bearing inserts* similar to those shown for the connecting rod in Fig. 12 are employed in the main bearings. Both main and connecting-rod bearings are lined with a good bearing material such as babbitt metal. The connecting-rod and main-bearing bores, inside and outside diameters of the inserts, and surfaces of the journals are all precision finished to sufficiently close tolerances so that a practically perfect running fit is obtained between the insert and the journal when they are assembled.

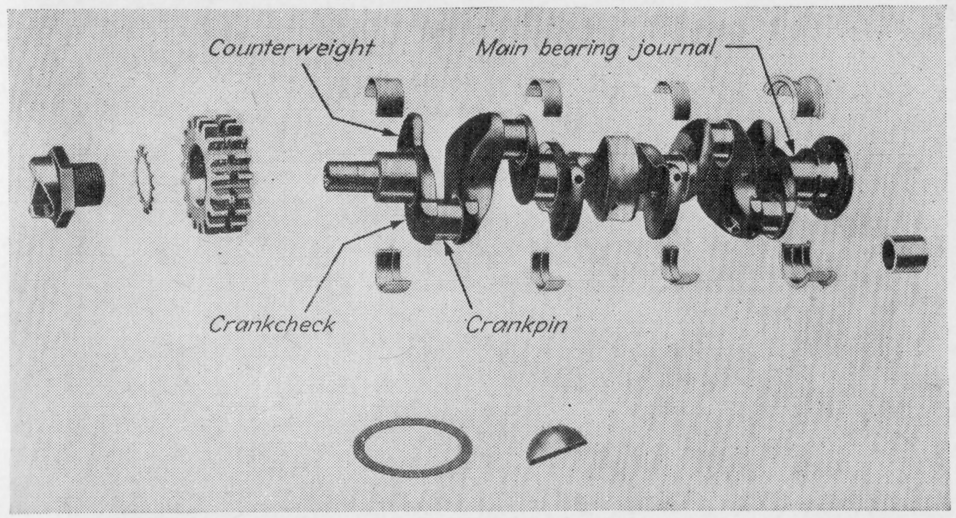

FIG. 11. Details of the crankshaft and some closely associated parts including the main-bearing inserts for the engine of Fig. 9. (*Chrysler Corp.*)

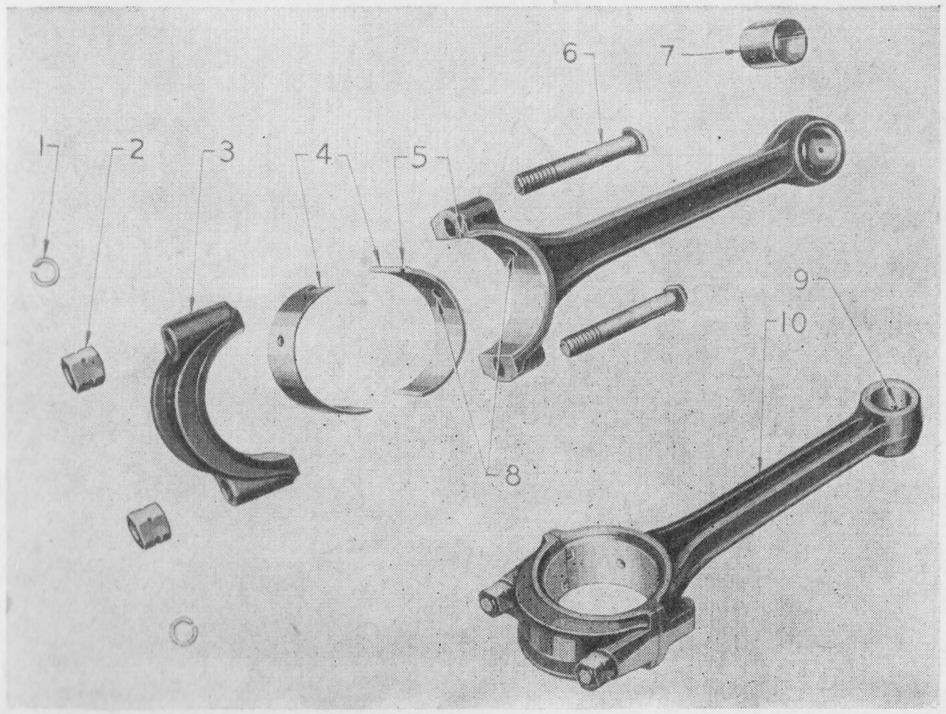

FIG. 12. Details of the connecting rod for the engine of Fig. 9. (*Chrysler Corp.*)

1. Cap bolt nut lock washer
2. Cap bolt nut
3. Cap
4. Rod bearings
5. Locating tongue and groove
6. Cap bolt
7. Piston pin bushing
8. Oil holes
9. Oil hole
10. Rod assembly

Figure 13 shows a piston together with a section through the *piston-ring belt.* The piston is an aluminum-alloy permanent mold casting slotted to allow for thermal expansion. The *piston pin* is prevented from floating axially in the piston by a wire *piston-pin retainer,* or *circlip,* made in the form of a horseshoe in such a way that it fits into a groove near the end of the piston-pin bore.

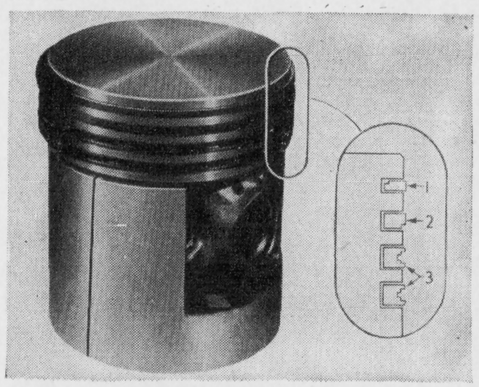

FIG. 13. Details of the piston and rings for the engine of Fig. 9. (*Chrysler Corp.*)

1. Top ring (compression ring)
2. Second ring (compression ring)
3. Oil scraper rings

The cylinders and their cooling jackets are usually made in the form of a single casting that includes the crankcase as well. The cylinder block for the engine of Fig. 9 was made in this way and is shown in Fig. 14, while the cylinder head is shown in Fig. 15. Both are of alloy cast iron with cooling water passages, intake and exhaust ports, and other details obtained in the casting by the extensive use of dry-sand cores. As many as 400 separate cores may be assembled in the mold for a cylinder block. A minimum amount of machining is

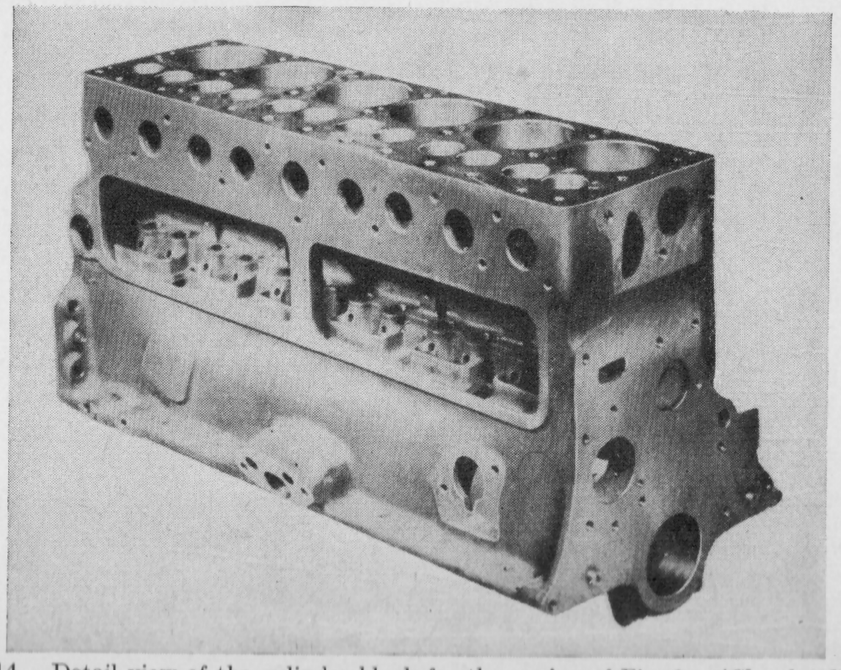

FIG. 14. Detail view of the cylinder block for the engine of Fig. 9. (*Chrysler Corp.*)

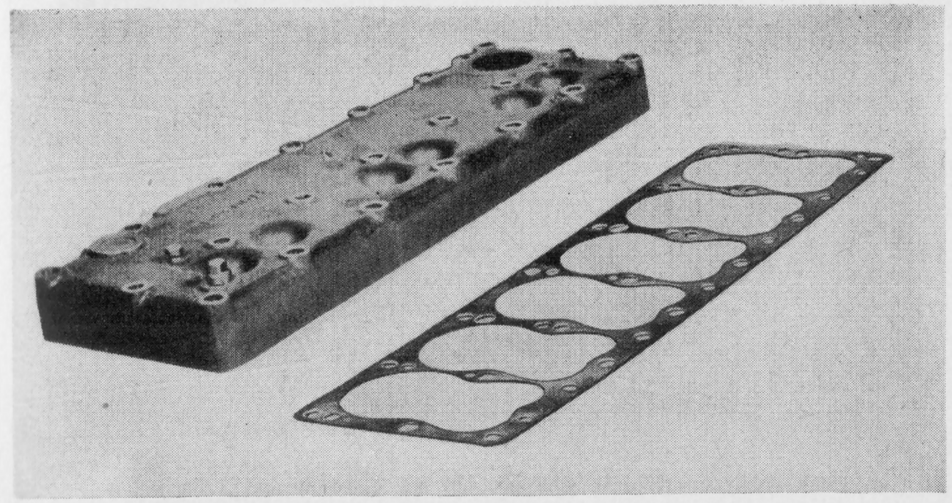

FIG. 15. Detail view of the cylinder head and cylinder-head gasket for the engine of Fig. 9. (*Chrysler Corp.*)

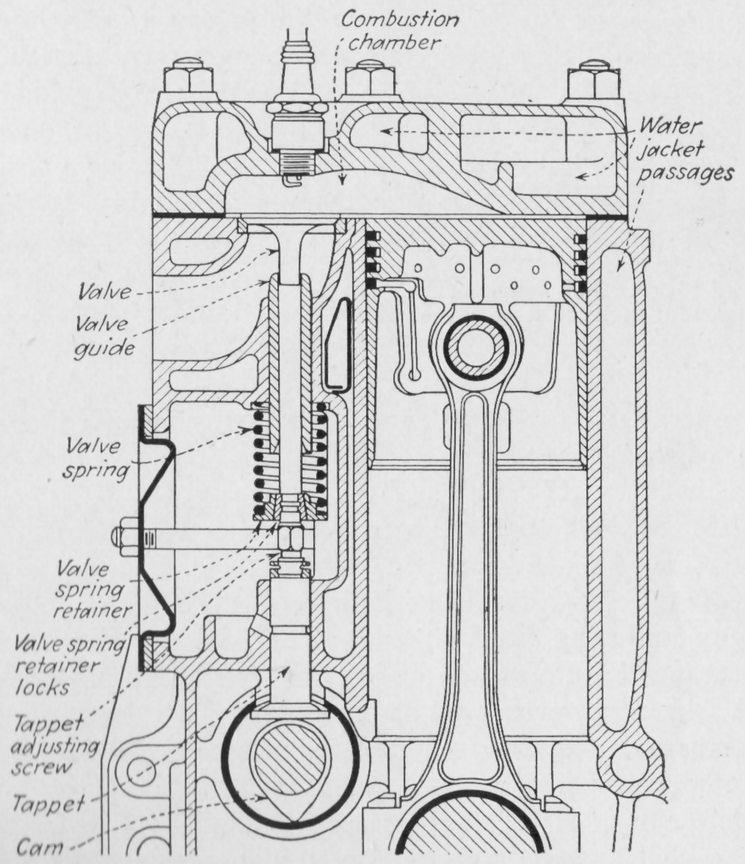

FIG. 16. Cross section through the combustion chamber and valve mechanism of the engine in Fig. 9. (*Chrysler Corp.*)

done to keep the cost low, but most of that which is necessary must be precise. The most important machine work is on the cylinders, which are bored and honed to close limits not only on the diameter but also on the straightness of bore and on the finish. The nature and quality of the finish are especially vital from the lubrication standpoint. The *valve-seat inserts* and *valve guides* (see Fig. 16) are pressed in and the studs "driven" (screwed in with a light interference fit) to give a *cylinder-block assembly.* The lower part of the crankcase is closed off by the *oil pan,* which may be seen in Fig. 9.

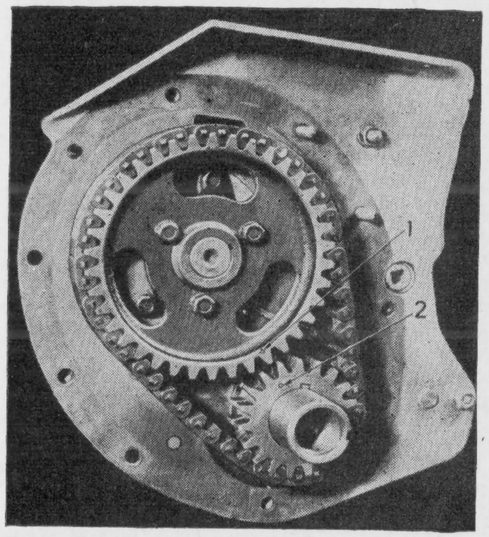

FIG. 17. Timing chain and sprockets for the engine of Fig. 9. The numerals in the diagram indicate: (1) Mark on camshaft sprocket; (2) mark on crankshaft sprocket. (*Chrysler Corp.*)

A large number of separate parts are used in the valve mechanism. The *poppet* valves reciprocate in *valve-stem guides,* or *valve guides.* The lower end of each *valve spring* bears against a *valve-spring washer,* or *valve-spring retainer,* which is restrained from slipping off the valve stem by *valve-spring-retainer locks,* or *valve locks.* As may be seen in Fig. 16, these have a conical outer surface to mate with the washer and a grooved inner surface to mate with the valve stem. They are split to permit assembly. Each valve is operated by a *tappet,* the upper end of which incorporates a *tappet adjusting screw,* which should be set to give a small clearance between the valve stem and tappet. This clearance is necessary because thermal expansion causes a greater increase in valve-stem length than in cylinder-block size when the engine heats up. As a result, in engines of this type the valve would be partly open during the compression and power strokes if sufficient "cold clearance" were not provided between the tappet and valve stem. A loss in power as well as burning of the valves would result if high-temperature gases were permitted to escape during combustion and the power stroke.

The lower ends of the tappets are actuated by *cam lobes* on the *camshaft,* which may be seen in Fig. 9. The camshaft is driven at one-half crankshaft speed by a *timing chain* and *timing-chain sprockets*

at the front end. These may be seen in Fig. 17 more clearly than in Fig. 9. Note the *timing marks* provided on the sprocket teeth to facilitate setting the valve timing during assembly. It is also fairly common practice to use a pair of plain or helical spur gears in place of

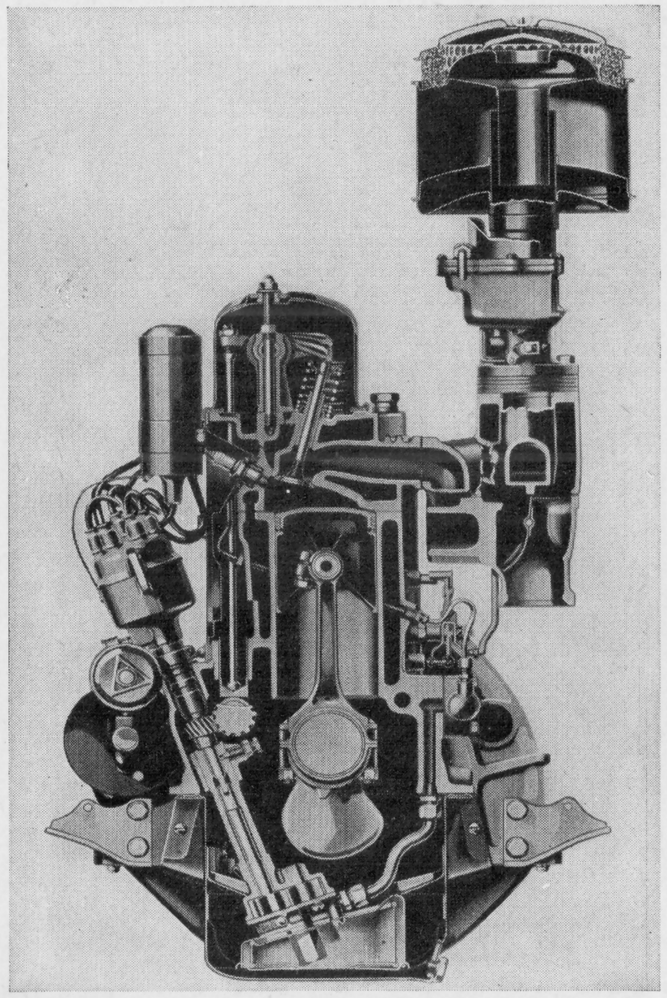

FIG. 18. Cross section through a six-cylinder overhead-valve Chevrolet engine. (*General Motors Corp.*)

a chain for the drive between the crankshaft and camshaft. If that construction is used, the camshaft gear is usually made of a composition material such as bakelite-impregnated fabric to inhibit vibration and noise. This is necessary because the valve mechanism imposes heavy intermittent loads, which reverse the direction of the load on the timing gear.

The *oil pump* and the *distributor* are driven from opposite ends of

the inclined shaft in Fig. 10. This shaft is driven from the camshaft at camshaft speed through a pair of spiral gears. The oil filter, the oil-pressure relief valve, the breather, and the oil strainer—which floats in the oil pan—may be seen in the same view. The intake and exhaust manifolds, the fan and water pump, and the generator pulley are shown in Fig. 9.

The Six-cylinder Overhead-valve Automobile Engine. Figure 18 is a cross-sectional view through an overhead-valve engine and shows the valve arrangement. Except for this item, such an engine is

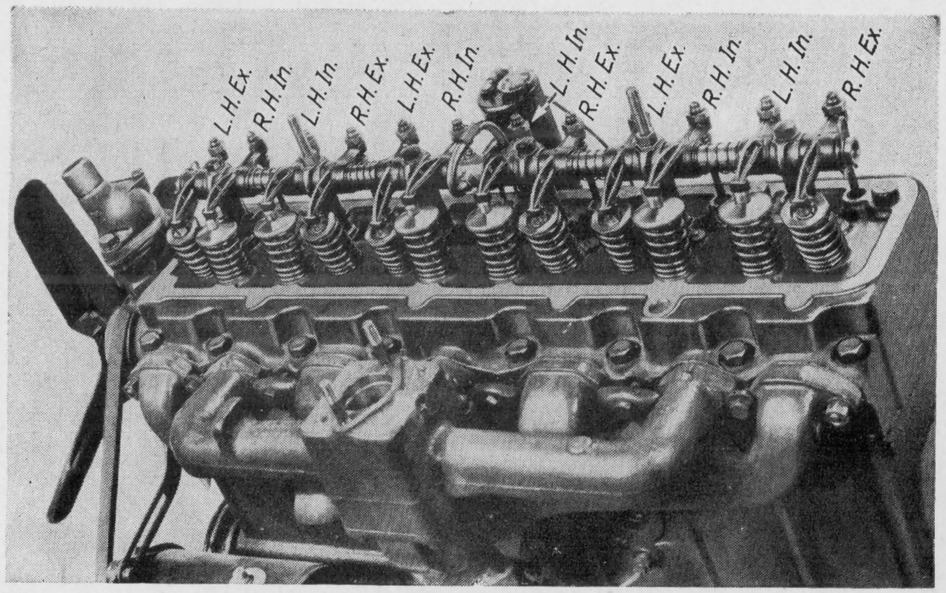

FIG. 19. Rocker arms, valves, and valve springs assembled on the engine of Fig. 18. (*General Motors Corp.*)

usually very similar in construction to the L-head engine just described. The section shown may be compared with a similar section through an L-head engine in Fig. 16. Figure 19 shows a view of the *valve gallery* as seen by looking down on the exterior of the head assembly with the valve cover removed. *Valve lifters* riding on the camshaft actuate the *push rods*. These operate the *rocker arms*, which, in turn, move the valves. An adjusting screw in one end of the rocker arm permits accurate setting of valve clearances. The longer column of steel between the cam and the valve head makes it necessary to use somewhat larger clearances than are required in L-head engines.

The Eight-cylinder V-engine. The eight-cylinder V L-head engine has proved to be a compact and widely used type of automotive power plant. Figure 20 shows one of these engines complete

with all its equipment, including the fuel pump and generator. Figure 21 shows the same engine with the fan, carburetor, ignition system, generator, fuel pump, and breather removed. An exploded view of the principal stationary parts is given in Fig. 22. One bank of cylinders is placed slightly ahead of the other so that side-by-side rods

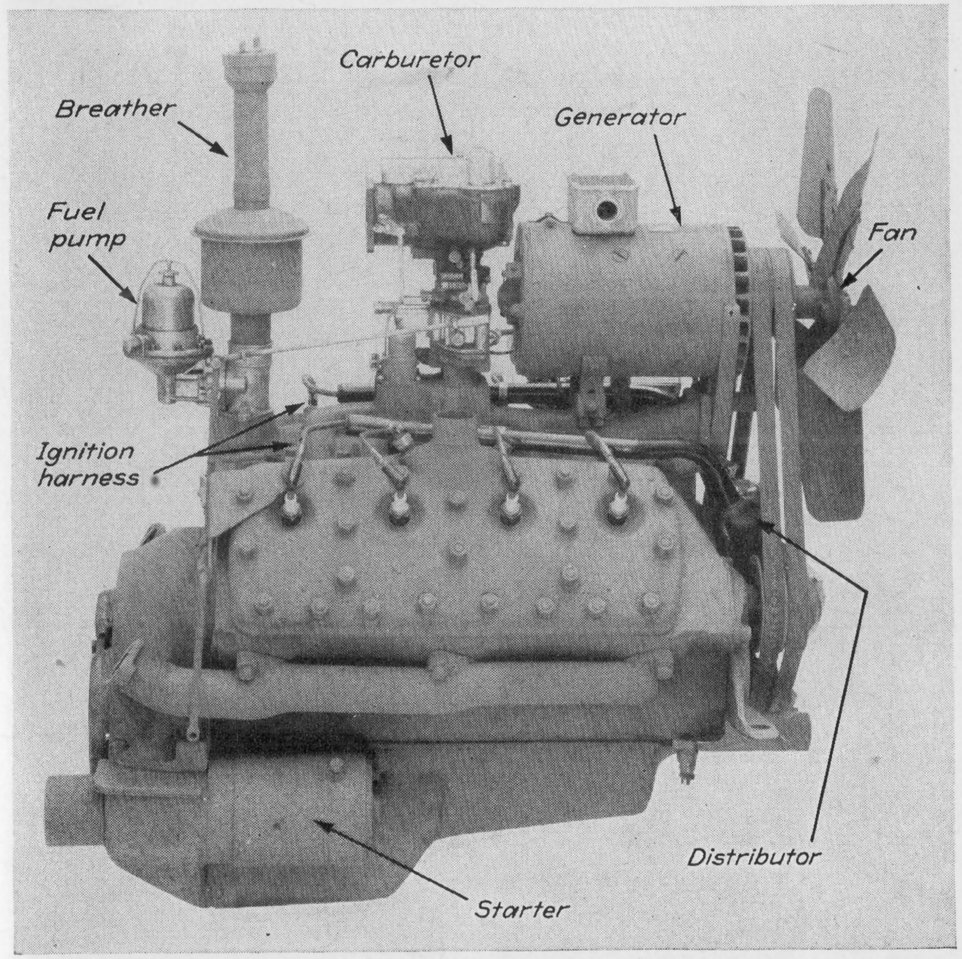

FIG. 20. Eight-cylinder Vee L-head passenger-car and truck engine. (*Ford Motor Co.*)

may be used, as is evident in Fig. 23, a view taken looking up into the crankcase with the pan removed. The floating oil screen, the three main bearings, and the counterweights on the crankshaft may also be seen in Fig. 23. The camshaft is located between the two cylinder banks, fitting into the bore in the center of the front of the cylinder block in Fig. 22. It is driven from the helical spur gear on the front of the crankshaft in Fig. 23. The distributor is mounted directly on the front of the camshaft, and may be seen in Fig. 20. It is centered

by a *pilot*, a circular section that protrudes from the distributor housing concentric with the shaft. This pilot fits into a shallow-bore concentric with the camshaft at the front of the cylinder block (see Fig. 21).

ENGINES FOR TRUCKS, BUSSES, AGRICULTURAL, AND EARTH-MOVING EQUIPMENT

Most engines in this category are very similar to the three engines described above. They differ principally in that as a group they are

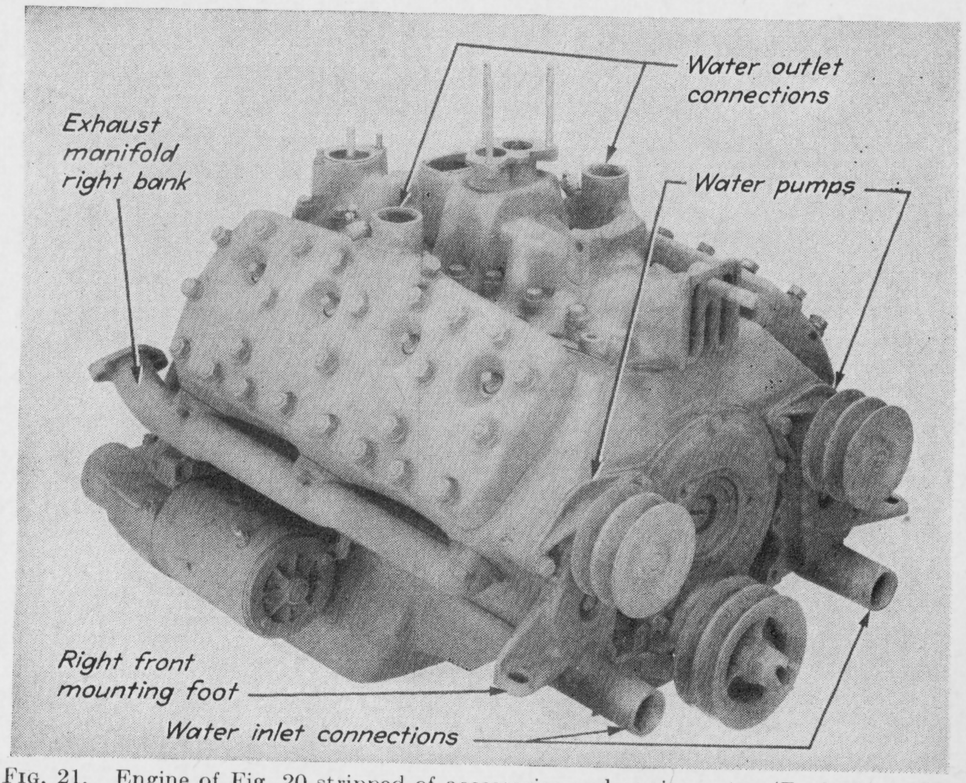

FIG. 21. Engine of Fig. 20 stripped of accessories and equipment. (*Ford Motor Co.*)

more ruggedly built and that many are diesels. The four-stroke-cycle diesel looks very much like a four-stroke-cycle spark-ignition engine, but the two-stroke-cycle diesel differs in a number of important respects.

The Two-stroke-cycle Automotive Diesel. Figure 24 shows a six-cylinder model of a two-stroke-cycle automotive diesel.[7] The construction is more massive than that of a passenger-car engine such as those in Figs. 9, 18, and 20, although this is a little difficult to see in

the photographs. The internal arrangement of the parts associated with the combustion chamber may be seen in Fig. 25. The Roots blower at the right of Fig. 25 supplies the intake port with scavenging air. The cycle follows the pattern described on page 6. The two exhaust valves in each cylinder head provide ample flow passage area

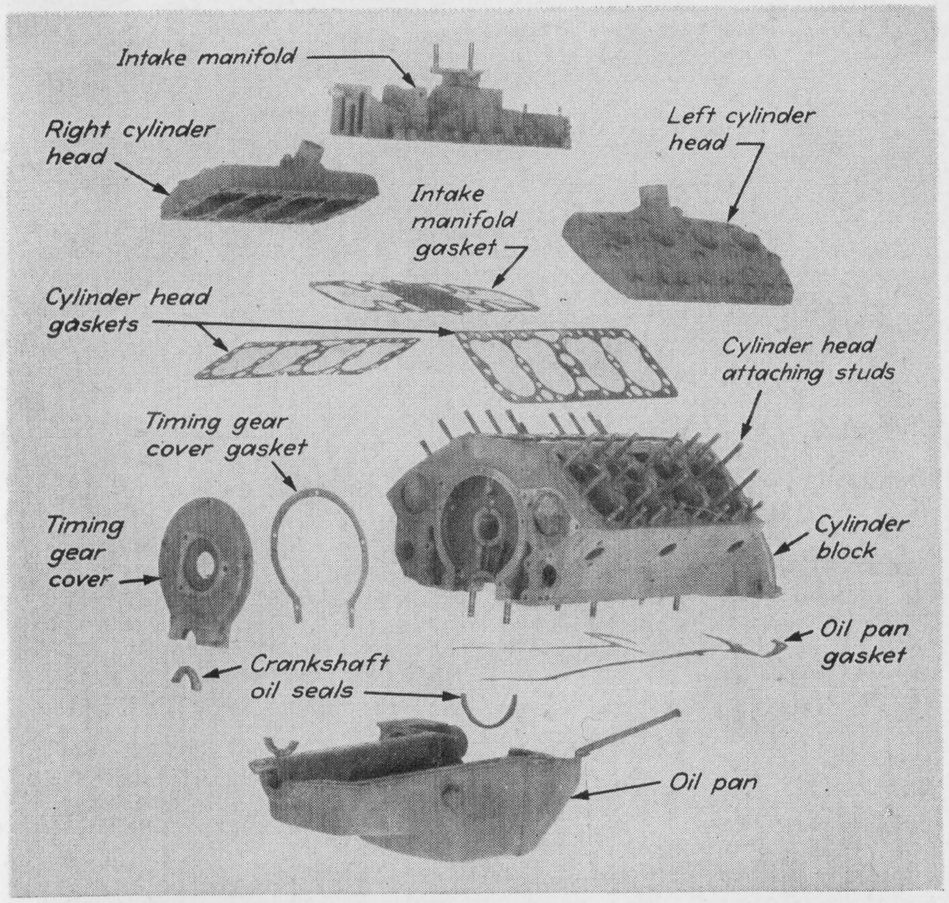

FIG. 22. Principal stationary parts in the engine of Fig. 20. (*Ford Motor Co.*)

for rapid scavenging. Both the valves and the unit fuel injectors are operated by rocker arms and short push rods from the camshaft, which runs at crankshaft speed. The dynamic unbalance inherent in the crank-throw arrangement best suited to this type of engine is compensated for by balance weights carried on a shaft on the opposite side of the cylinders from the camshaft but at the same height above the crankshaft. These two shafts are driven by the crankshaft through a gear train at the end of the cylinder block.

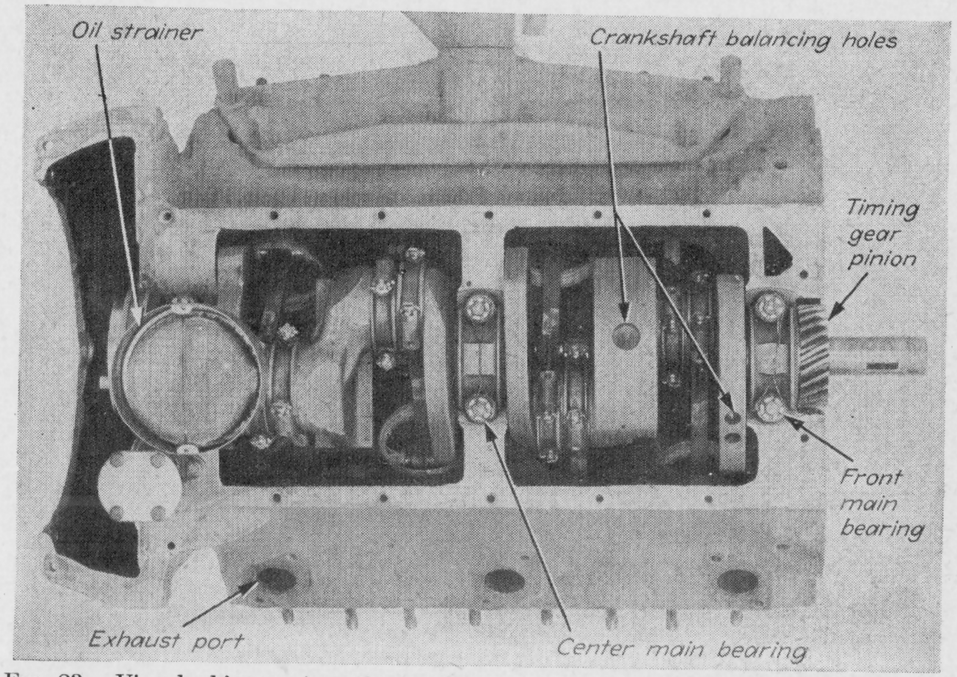

FIG. 23. View looking up into the crankcase of the engine of Fig. 20 with the oil pan removed. (*Ford Motor Co.*)

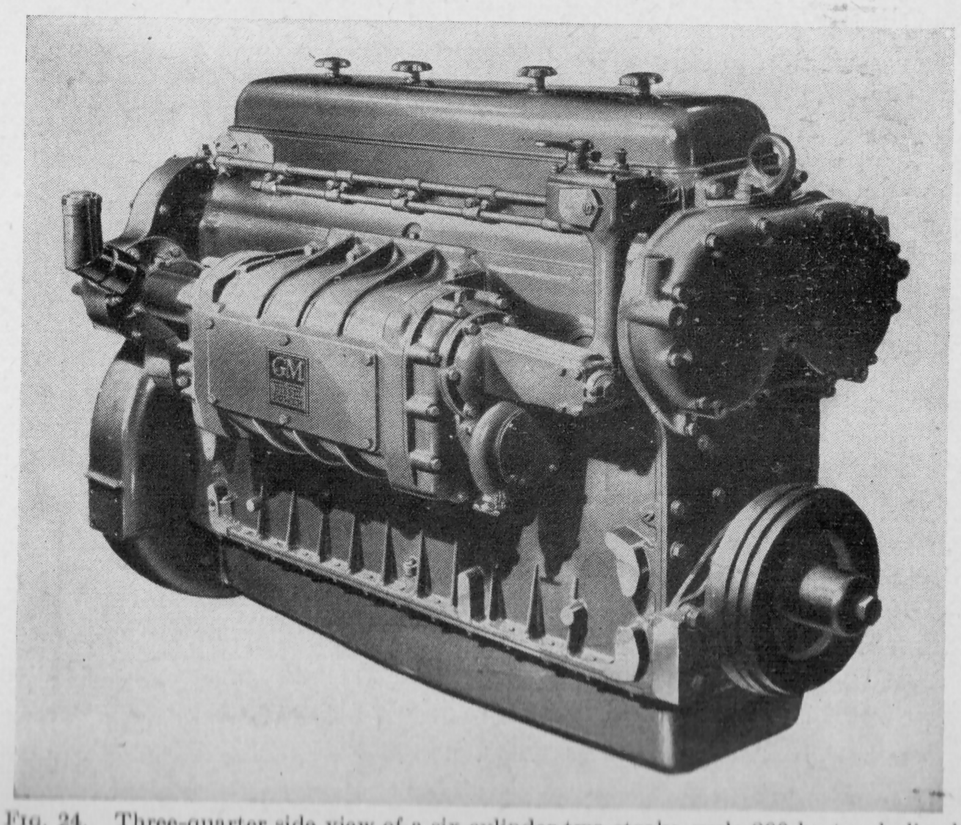

FIG. 24. Three-quarter side view of a six-cylinder two-stroke-cycle 200-hp truck diesel engine. (*Detroit Diesel Division, General Motors Corp.*)

Engines essentially similar to that described have been built in 3-, 4-, 12-, and 24- as well as in 6-cylinder units. The two larger sizes are made by coupling together two or four 6-cylinder units. These multiple-engine power plants have seen extensive military use as primary power plants both in naval landing craft of all sizes, up to and

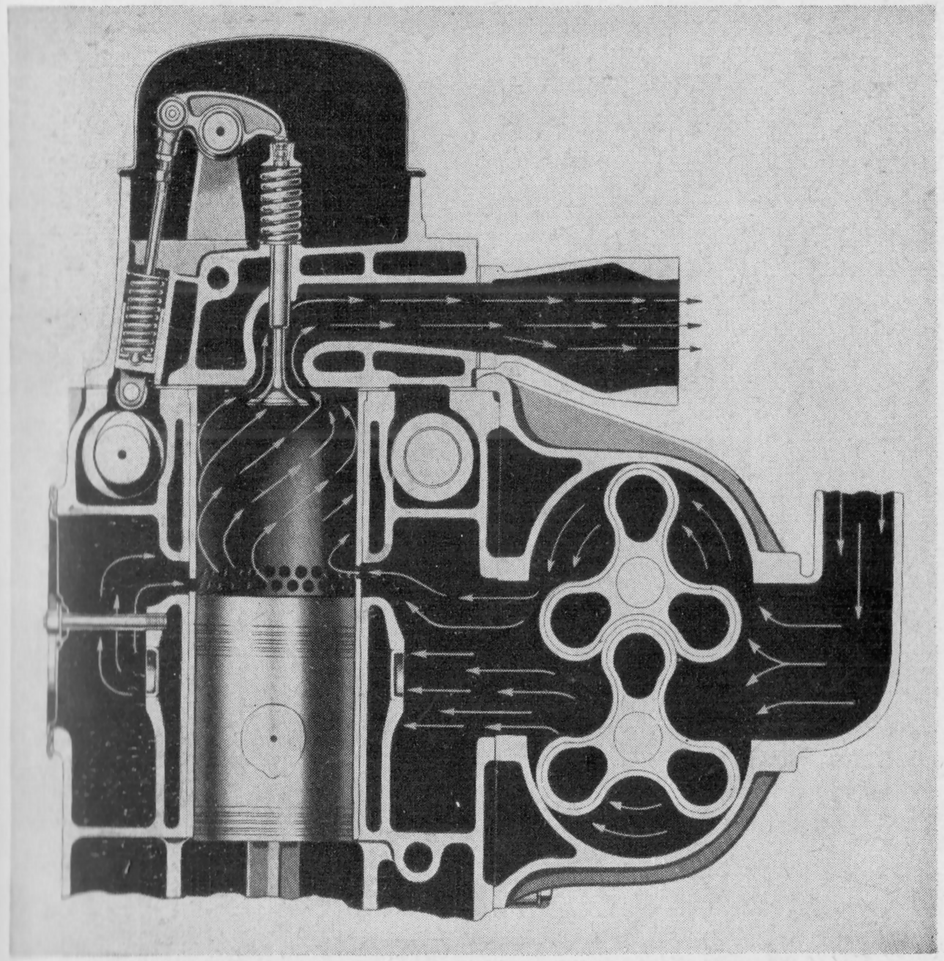

FIG. 25. Cross section through the engine of Fig. 24. (*Detroit Diesel Division, General Motors Corp.*)

including the LST class of ship, and in army medium tanks. The four- and six-cylinder engines have been widely used in trucks, busses, tractors, and bulldozers and, with electrical generators, in portable power plants.

UNSUPERCHARGED AIRCRAFT ENGINES

Practically all aircraft engines developing less than 200 hp are unsupercharged and air-cooled. Five-, seven-, and nine-cylinder radial,

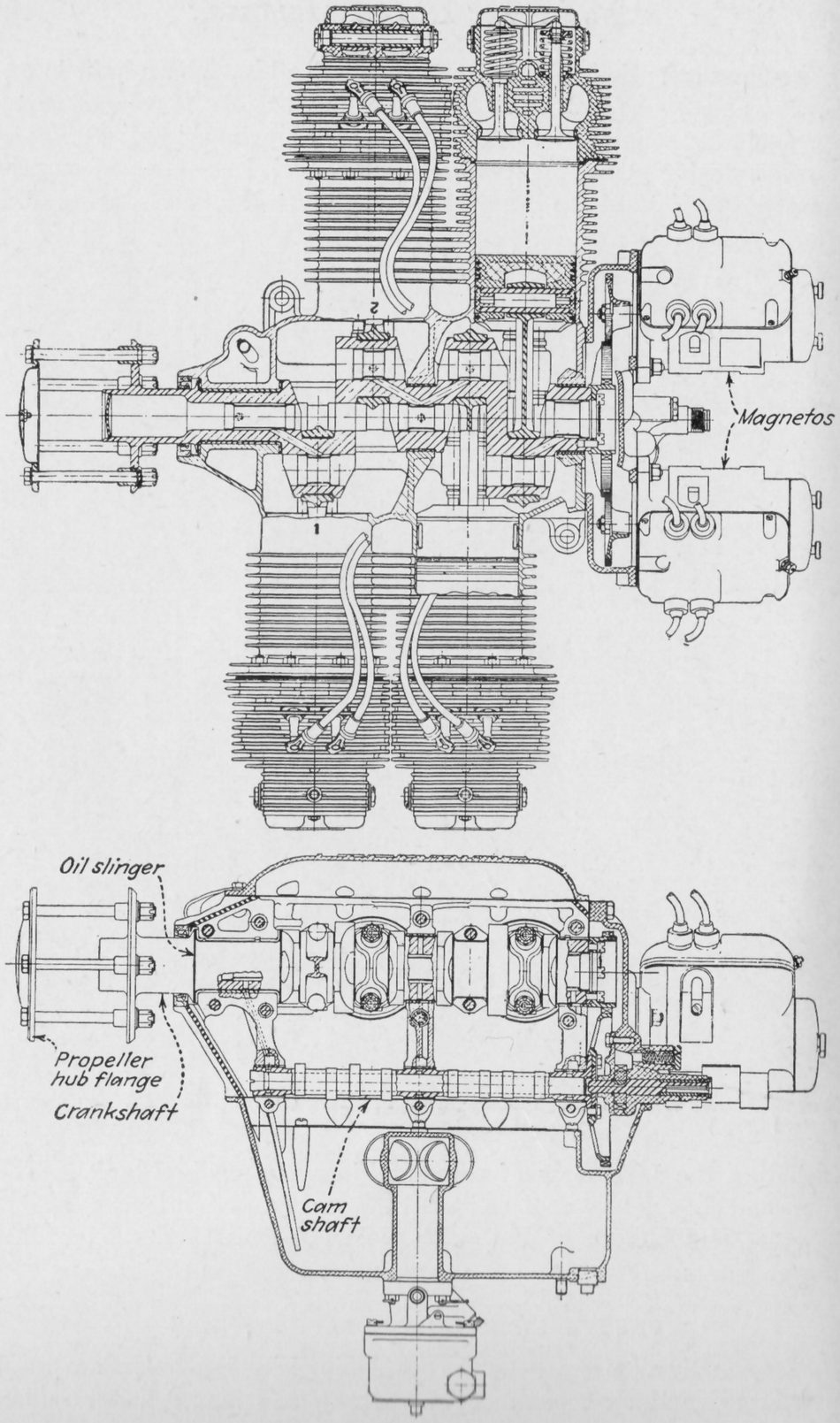

FIG. 26. For descriptive legend see opposite page.

four- and six-cylinder in-line, and four-, six-, and eight-cylinder horizontally opposed engines, are built by over a dozen manufacturers.[8] The cylinder construction, the valve gear, and the provisions for cooling are much the same in all these engines. The horizontally opposed cylinder engine, the most widely used type, is similar in construction to the in-line engines. The small radial engines are built in much the same fashion as their large supercharged counterparts (see F. H. Colvin[9]).

Horizontally Opposed Light-aircraft Engines. The engine shown in Fig. 26 is representative of the four- and six-cylinder models that power the majority of aircraft employing engines of 150 hp or less. Many of the parts are made and finished in the same way as parts made for passenger cars by the same vendors. Spark plugs, valves, valve springs, camshafts, crankshafts, connecting rods, and main and connecting-rod bearings, as well as most of the smaller parts, are

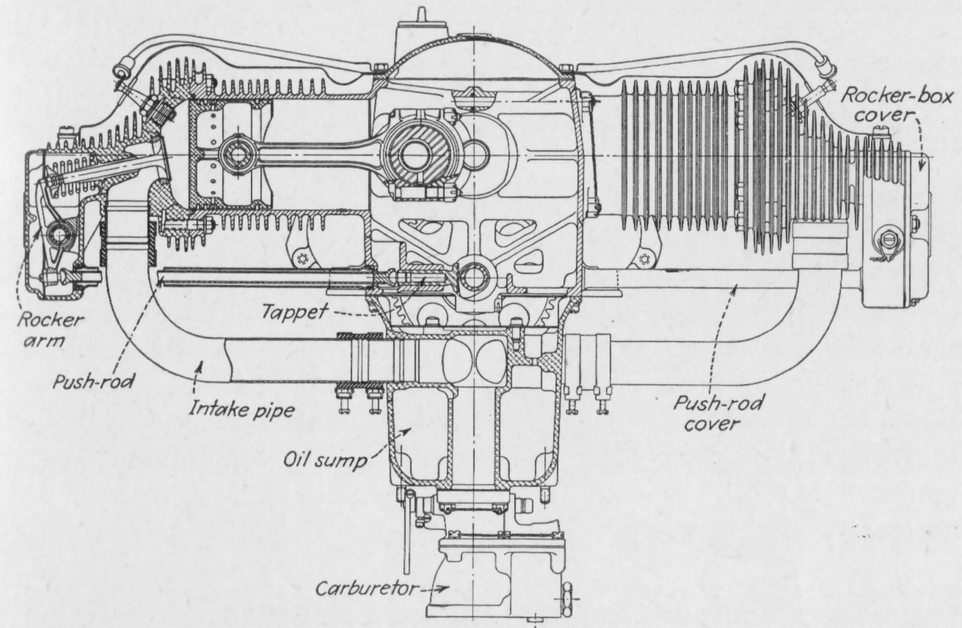

FIG. 26. Three-view drawing of a 100-hp four-cylinder horizontally opposed aircraft engine.

almost indistinguishable from their automobile counterparts. The principal differences lie in the crankcase and the cylinders. The cast-aluminum crankcase is split longitudinally along the vertical center line. The *oil sump*, also of cast aluminum, is mounted on the bottom of the crankcase. It can be seen in Fig. 26 that a part of the induction system is cast integral with the oil sump, partly to give

some oil cooling but primarily to heat the fuel-air charge as an aid to fuel vaporization. The carburetor is mounted on the bottom of the oil sump, while *intake pipes* carry the charge from the side of the oil sump to the cylinders. The four-throw one-piece forged-steel crankshaft is carried in four main bearings. The propeller thrust is transmitted from the shoulder on the front crankcheek just behind the second main bearing to the faced plain-bearing surface. Some six-cylinder models employ ball bearings in place of the plain main bearing immediately behind the propeller, in which case the ball bearing carries the thrust. The *oil slinger* at the front of the crankshaft is a circular fin made integral with the shaft in this case. Oil tending to creep forward along the shaft is thrown off by centrifugal force as it passes over the outer edge of this fin, thus greatly reducing the load on the oil seal just to the left of the slinger. The overhead valves are actuated by push rods and rocker arms from the camshaft, which is carried in the crankcase just above the crankshaft. The cylinders have a cast-aluminum head screwed and shrunk onto a steel barrel having integral machined cooling fins. Figure 27 shows one of these cylinders. Note that a bronze bushing is provided to receive the spark plug because threads in aluminum have a tendency to seize and tear if repeated installation and removal are necessary.

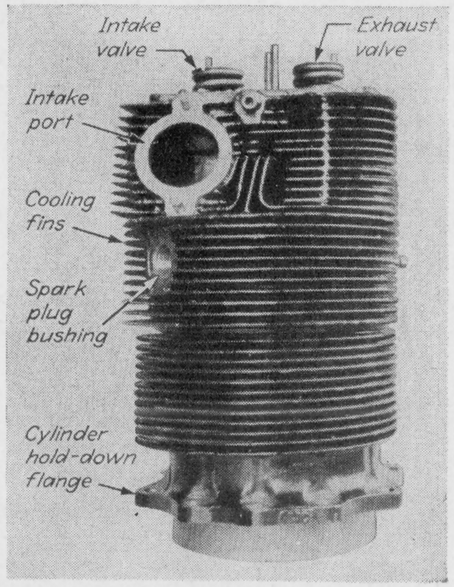

FIG. 27. Cylinder-head and barrel assembly for an engine similar to that of Fig. 26.

SUPERCHARGED AIRCRAFT ENGINES

Air-cooled Radial Engines. A partly cut away model of a representative supercharged single-row radial air-cooled aircraft engine is shown in Fig. 28, while Fig. 29 shows a longitudinal section through the same engine. A single-throw fully counterweighted crankshaft is mounted in roller bearings in the main crankcase. An *articulating-rod system* is employed to connect all nine pistons to a common crankpin. The *articulating rods* are connected to the *master rod* by means of *knuckle pins* as shown in Fig. 30. The cylinders have cast-aluminum heads screwed and shrunk on machined steel barrels in much the same way as the cylinder in Fig. 27. The valves are operated through

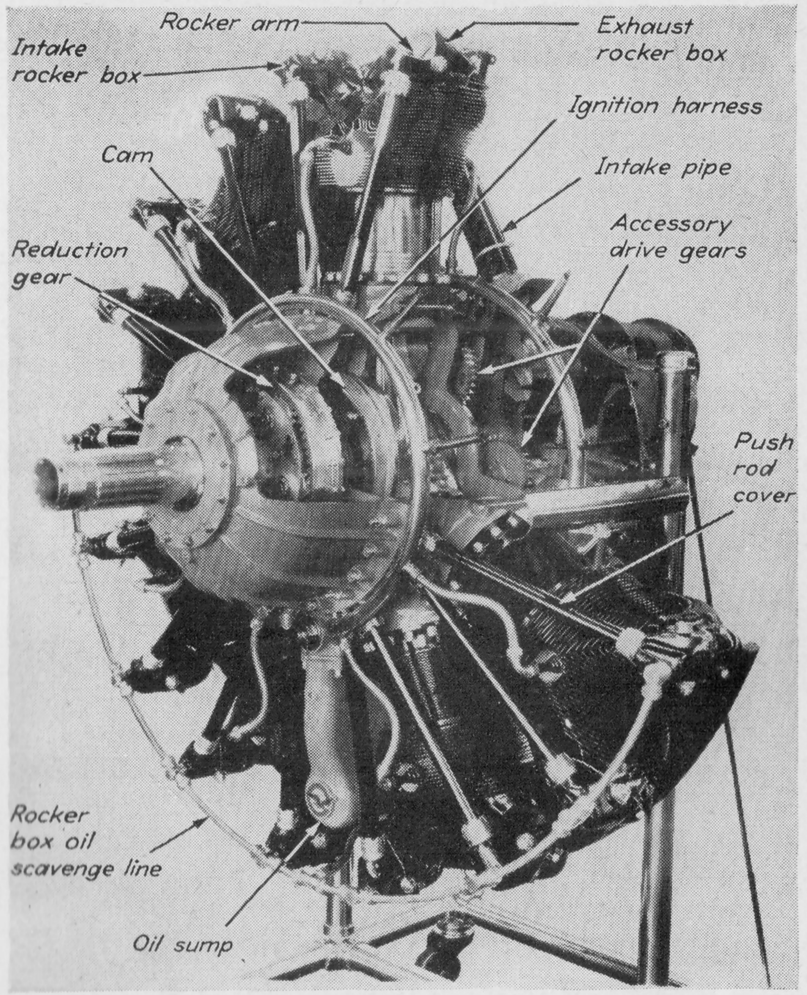

FIG. 28. Nine-cylinder single-row radial air-cooled aircraft engine that has been partially cut away to show the interior arrangement of the parts. (*Pratt & Whitney Aircraft.*)

rocker arms, push rods, and tappets by a large ring-type cam mounted concentric with the crankshaft just back of the planetary propeller-shaft reduction gear. These parts may be seen in both Figs. 28 and 29. The four-lobed cam rotates at one-eighth crankshaft speed. Two parallel sets of lobes are provided—one for the intake and one for the exhaust valves. The supercharger impeller is mounted on a

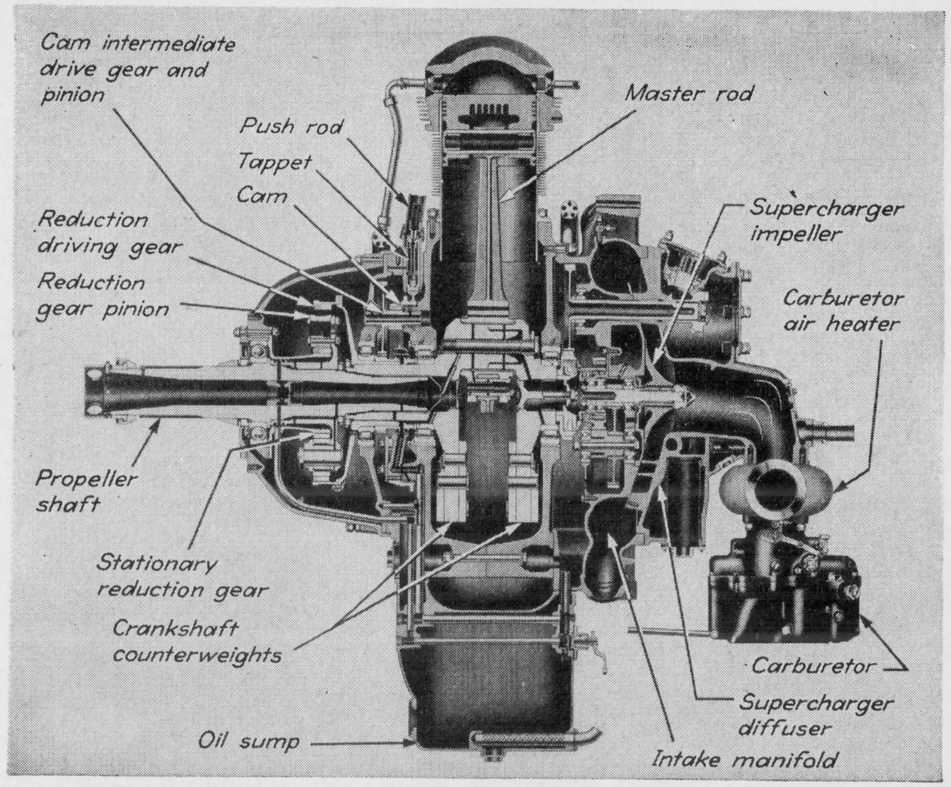

FIG. 29. Longitudinal section through the engine of Fig. 28. (*Pratt & Whitney Aircraft.*)

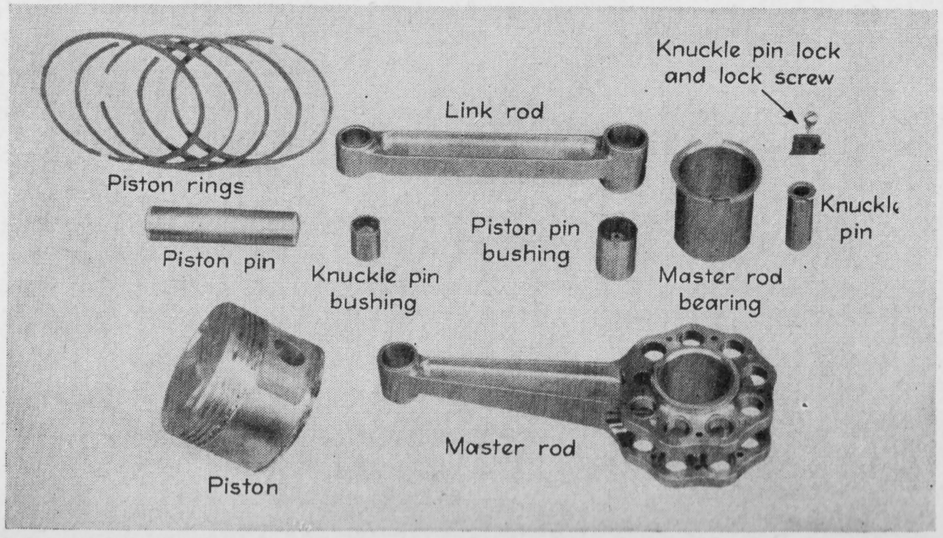

FIG. 30. The master rod, a link rod, and associated detail parts for the radial engine of Fig. 28. Note that the large load-carrying bearing sleeves are called *bearings*, while the smaller ones are called *bushings*. (*Pratt & Whitney Aircraft.*)

shaft concentric with the crankshaft but is driven through a lay shaft at five to ten times crankshaft speed depending on the engine model. The fuel-air charge passes through the supercharger to an annular intake manifold, from which it is carried to the individual cylinders by intake pipes. The engine is supported by means of *mounting lugs* cast integral with the front supercharger housing between the intake-pipe bosses. Most of the accessories are mounted on the supercharger housing. They are driven by accessory drive shafts, which in turn are driven from the main accessory drive gear through spur and/or bevel gears.

Large Stationary, Marine, and Railroad Diesel Engines. Both two- and four-stroke-cycle diesel engines are widely used in the stationary, marine, and railroad fields. Four-cycle spark-ignition gas engines similar in size and construction are used extensively in stationary applications where a low-cost gas supply is readily available. In some stationary diesel engines, provision has been made so that it is possible to operate on either diesel fuel oil alone or both fuel oil and gas. In the latter case a lean mixture of gas is employed, and a small amount of diesel fuel is injected to serve as a means of ignition. Very few gasoline engines are employed in these fields of large heavy-duty power plants, although 1,500-hp spark-ignition gasoline engines are used in motor torpedo and racing boats, where weight is a vital consideration.

V-16 *Marine Diesel.* The 1,600-hp engine shown in Fig. 31 is representative of the large diesel engines built for stationary, marine, and railroad service. It is a two-stroke-cycle engine much like the truck diesel engine in Fig. 24. In fact, the design of that truck diesel engine was based on an earlier model of the much larger marine engine in Fig. 31. Fuel-injection and combustion troubles to be discussed in Chaps. IV and VI delayed the development of small high-speed diesel engines until ten or twenty years after widespread commercial use of large stationary and marine diesel engines.

An inspection of Fig. 31 and the vertical cross section of the same engine in Fig. 32 shows that, except for the larger size, most of the details of construction are similar to those of the engines described previously. One exception is the starting air manifold, which, coupled with the necessary valves in the cylinder heads, is common only in the stationary-, marine-, and railroad-engine types. The torque required to turn one of these large engines over is so great that it has been found easiest to start them with compressed air, a supply of which is usually available or can be easily provided from a small

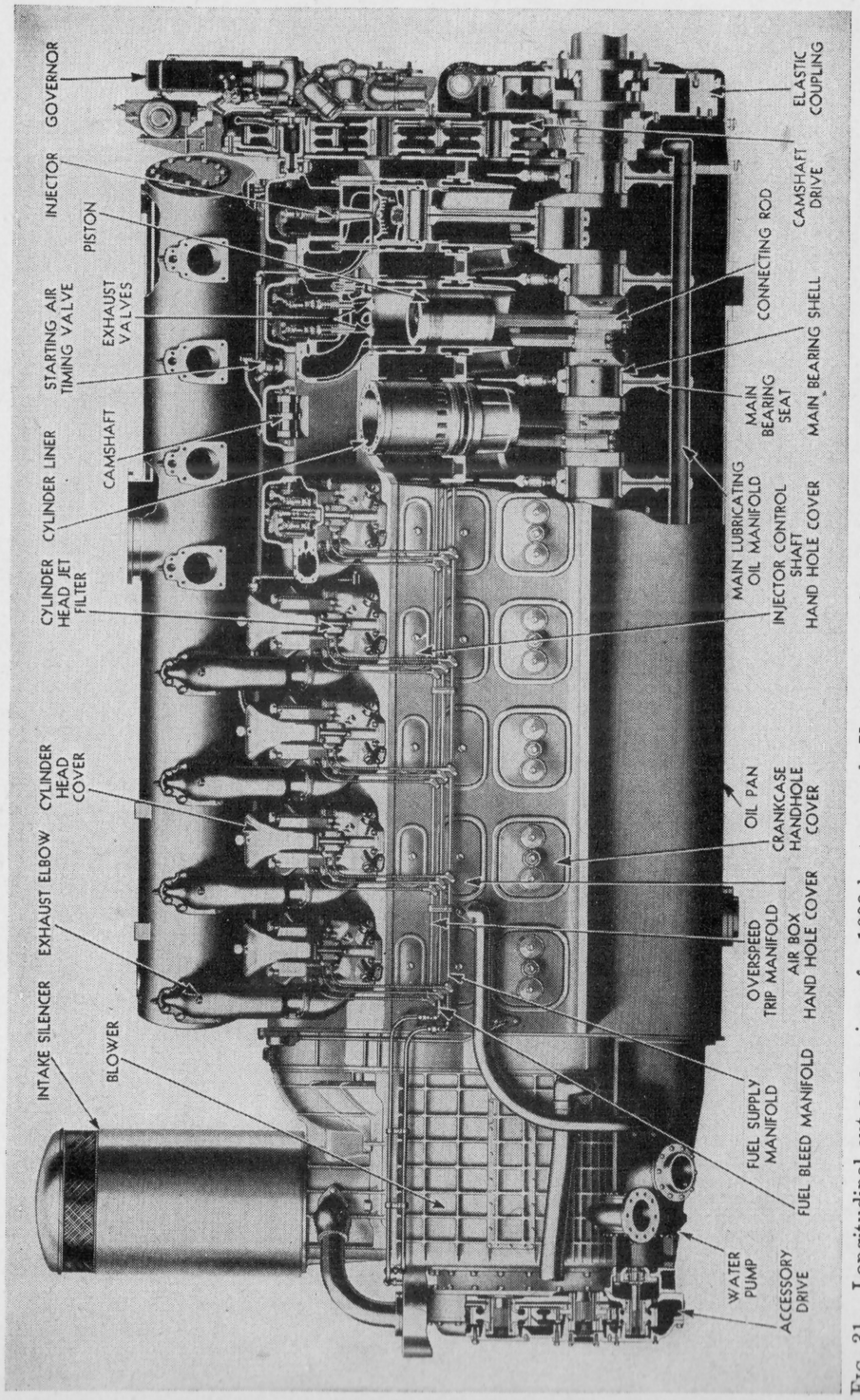

FIG. 31. Longitudinal cut-away view of a 1600-hp two-cycle V-16 marine diesel engine. (*Cleveland Diesel Division, General Motors Corp.*)

compressor and storage tank. The extensive use of handhole covers to facilitate inspection and maintenance (often while the engine is running) are features peculiar to the large engines. Note, too, in Fig. 31 that a thermocouple is mounted in each exhaust port so that

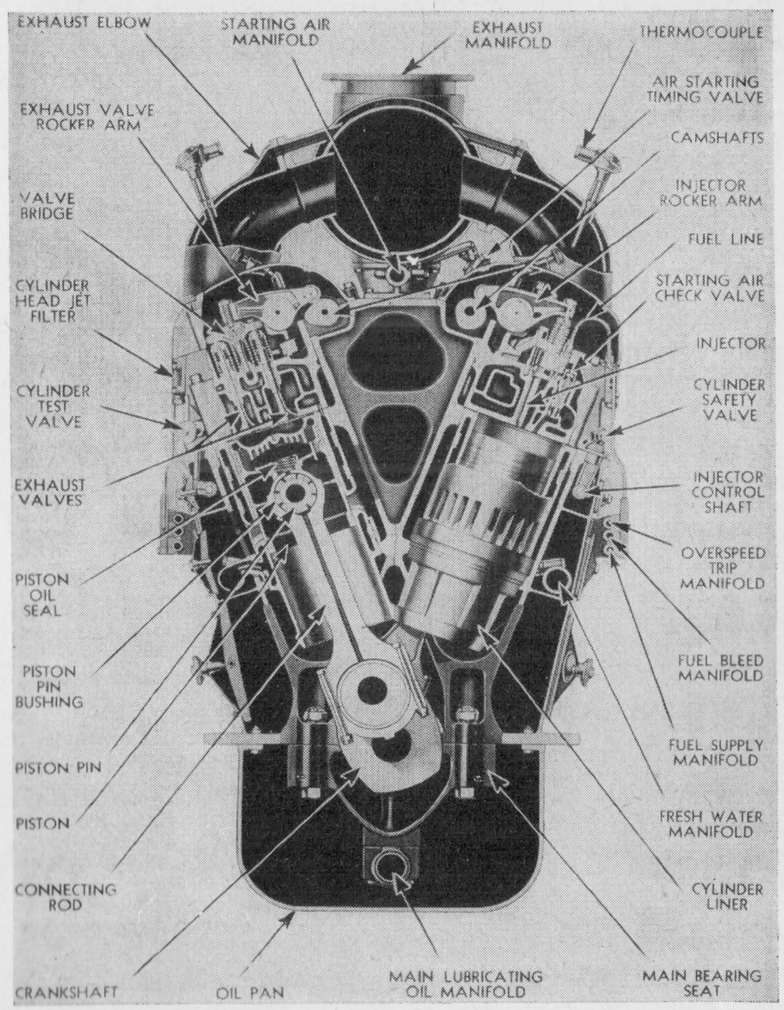

FIG. 32. Cross-sectional view through the engine of Fig. 31. (*Cleveland Diesel Division General Motors Corp.*)

the exhaust temperature of each cylinder can be checked readily. This is just another indication of the differences between automotive and stationary, marine, and railroad service. In the latter groups an engine operator frequently checks over the engine while it is running to catch any indications of trouble and keep the exterior of the engine bright and clean. Automotive engines never get that sustained,

close attention because the size and cost of the engine do not justify it and the type of service does not place such a heavy penalty on an unexpected shutdown, however brief, as would be entailed in the case of electric power plant, ship, or locomotive. It is very difficult to approach perfection, and it is thus understandable that engines in these heavy-duty applications are usually about ten times as heavy and cost about ten times as much per horsepower as automotive engines because their reliability and service life have been increased proportionately.

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Problems

1. In what respects do motor-vehicle-engine requirements differ from those for a stationary engine in an electric power plant?

2. At what speed should the camshaft of a four-stroke-cycle engine run? A two-stroke-cycle engine? Give answers in terms of crankshaft speed.

3. What is meant by the term "piston displacement"? In what units is it usually given?

4. Determine the piston displacement of a six-cylinder passenger-car engine having a $3\frac{1}{2}$-in. bore and a $3\frac{3}{4}$-in. stroke.

5. List seven distinguishing features of any given engine such as air or liquid cooling, etc.

6. Distinguish between the following:

Bushing and bearing
Crank throw and crankcheek
Pilot and journal
Intake port and intake manifold
Cylinder head and cylinder block
Push rod and connecting rod
Gasket, spacer, and shim
Valve gallery and crankcase
Tappet and valve guide

CHAPTER II

IDEALIZED CYCLES

The increasing cost and complexity of engine test work in recent years has made the problems of engine-cycle analysis important to every engineer in the field. This chapter reviews some principles of elementary thermodynamics and extends them toward the more specialized problems of combustion engines. The theoretical treatment is not simply of academic importance—it has many eminently practical applications. As will be demonstrated in Chap. V, Cycle Analysis, there is no appreciable gap between "theory" and "practice"—if the theory includes all the important factors that operate in practice.

The Constant-volume Cycle. Of the three basic thermodynamic cycles with which this text is primarily concerned, by far the most widely used is the constant-volume cycle. The well-known Otto cycle is the particular form of constant-volume cycle that is carried out in a four-stroke-cycle spark-ignition engine. Because the majority of internal-combustion engines are of the latter type, reference is often made only to the Otto cycle. However, other forms of the constant-volume cycle such as those in many compression-ignition and two-stroke-cycle engines are also of great practical importance.

The Air-standard Constant-volume Cycle. A first (and very rough) approximation for the operating cycle of a modern high-speed combustion engine is given by what is commonly referred to as the *air-standard cycle*. This is most conveniently described by the familiar pressure-volume ($P - V$) diagram shown in Fig. 33. Air is compressed adiabatically from standard atmospheric conditions at point 1 to a greatly reduced volume at point 2. Heat is then added to cause an increase in pressure at constant volume to point 3. This is followed by an adiabatic expansion to point 4, after which the working charge is allowed to exhaust to atmosphere with a constant cylinder volume until the pressure drops to atmospheric. The piston then moves upward to push out the gas remaining in the cylinder and moves back downward to draw in a fresh charge. The latter action gives a constant-pressure line across the bottom of the diagram.

The power delivered and the over-all thermal efficiency are probably the two most important characteristics of a cycle of this sort. Expressions for these quantities can be easily derived from simple

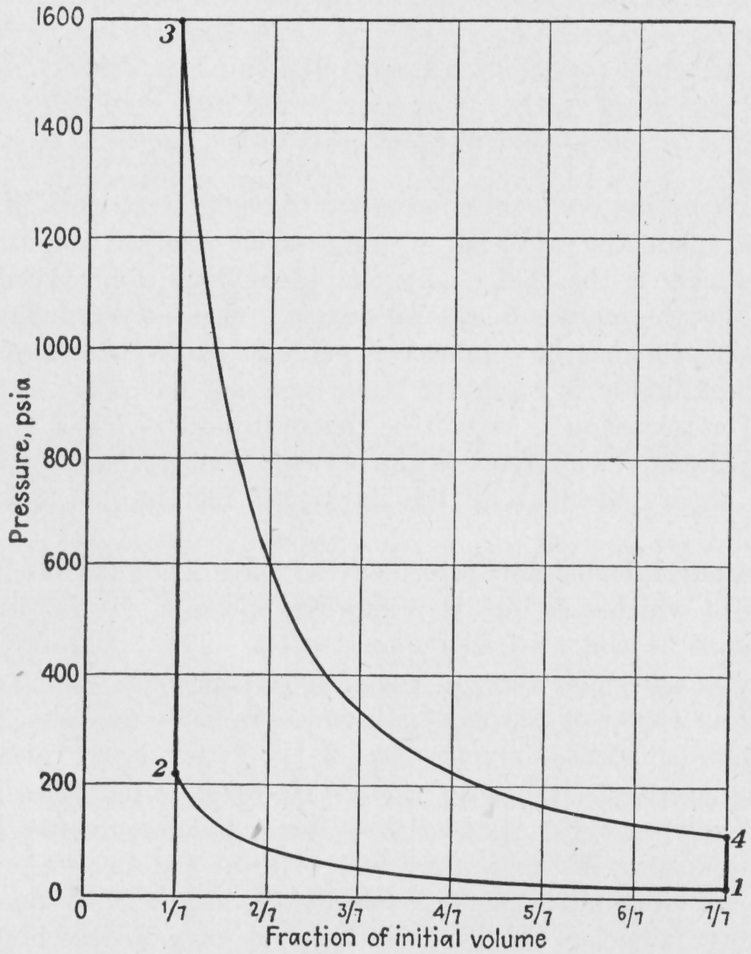

FIG. 33. Pressure-volume diagram for a constant-volume Otto cycle employing an idealized working fluid having the properties of air at room temperature. The compression ratio was taken as 7: 1 and the heat added as that from the complete combustion of a chemically correct mixture of isooctane and air.

thermodynamic relations. The conventional symbols for the various quantities involved in the derivations are defined as follows:

- P pressure, lb/ft²
- V volume, ft³
- v specific volume, ft³/lb
- T temperature, °R
- R gas constant, ft-lb/(lb)°(R)
- W weight, lb

- c_p specific heat at constant pressure, Btu/(lb)(°F)
- c_v specific heat at constant volume, Btu/(lb)(°F)
- k ratio of the specific heats, c_p/c_v
- J mechanical equivalent of heat, ft-lb/Btu
- Q quantity of heat added, Btu
- η_{th} thermal efficiency

With the above notation and numerical subscripts to denote points corresponding to those in Fig. 33, the work input from the piston to the cylinder charge of weight W is

$$\text{Work input} = Wc_v(T_2 - T_1)$$

Similarly, the work output from the cylinder charge to the piston and the resulting net work become

$$\text{Work output} = Wc_v(T_3 - T_4)$$
$$\text{Net work} = Wc_v(T_3 - T_4 - T_2 + T_1)$$

The heat energy Q put into the cycle may be related to these quantities by

$$Q = Wc_v(T_3 - T_2)$$

The thermal efficiency η_{th} of the cycle thus becomes

$$\eta_{th} = \frac{\text{net work}}{Q} = \frac{Wc_v(T_3 - T_4 - T_2 + T_1)}{Wc_v(T_3 - T_2)}$$
$$= \frac{T_3 - T_2}{T_3 - T_2} - \frac{T_4 - T_1}{T_3 - T_2}$$

But

$$T_3 = T_4\left(\frac{V_1}{V_2}\right)^{k-1} \qquad \text{and} \qquad T_2 = T_1\left(\frac{V_1}{V_2}\right)^{k-1}$$

Hence

$$\eta_{th} = 1 - \frac{T_4 - T_1}{T_4(V_1/V_2)^{k-1} - T_1(V_1/V_2)^{k-1}}$$
$$= 1 - \frac{1}{(V_1/V_2)^{k-1}} \qquad (1)$$

This relation can be used to give a different expression for the work obtained from the cycle.

$$\text{Net work} = Q\eta_{th} = Q\left[1 - \frac{1}{(V_1/V_2)^{k-1}}\right] \qquad (2)$$

Equations (1) and (2) show that the thermal efficiency is a function only of the compression ratio and the ratio of the specific heats k of the fluid used in the cycle, while the power output is a function of both these factors as well as the amount of heat added during each cycle.

Approximations to Actual Cycles. Equations (1) and (2) may be applied to give approximations to the actual constant-volume cycle in several ways. The simplest approximation is the air-standard cycle, in which the specific heats at constant pressure and constant volume of the working fluid are taken to be equal to the values for air at standard conditions and the heat added is considered to be that available from the chemically correct amount of a typical fuel required to react with all the oxygen in the air. The fluid at the beginning of the cycle is most often taken as being at standard sea-level atmospheric conditions, *viz.* 60°F and 29.92 in. Hg abs. A diagram for this hypothetical cycle is shown in Fig. 33.

A second approximation to the actual cycle can be made by employing air as a working fluid and allowing for the fact that the specific heats of air increase with temperature.[1] Figure 34 shows the effect of temperature on the specific heat at constant pressure for air and several other gases.[2] If allowance is made for the variable specific heats of air, it will be found that they have two major effects. As is evident from the preceding derivation, when both the heat added per pound of mixture and the temperature at the end of the compression stroke, T_2, are fixed, an increase in the specific heat must bring about a reduction in peak temperature. At the same time, since the difference between the specific heats of a gas is constant, an increase in specific heat must mean a reduction in k, the ratio of the specific heats. It is evident from Eqs. (1) and (2) that the latter will reduce the thermal efficiency and hence the useful work.

A third approximation to the actual cycle can be made by allowing for the fact that a part of the working fluid consists of carbon dioxide and water vapor after combustion. Figure 34 shows that these compounds have higher specific heats than air. This factor thus tends to increase further the average specific heats of the mixture, causing a further reduction in k, and hence a reduction in both power output and efficiency.

A fourth approximation to the actual cycle can be made by taking into consideration yet another factor, this one chemical in nature. At high temperatures the carbon dioxide and water-vapor molecules tend to become unstable so that small percentages of them break up, or dissociate, into carbon monoxide, oxygen, hydrogen, and OH ions.

Small amounts of nitrous oxide and atomic hydrogen, oxygen, and nitrogen are also formed. These reactions are endothermic and absorb a part of the heat that would otherwise be added by combustion. While most of the free ions combine as the temperature drops during

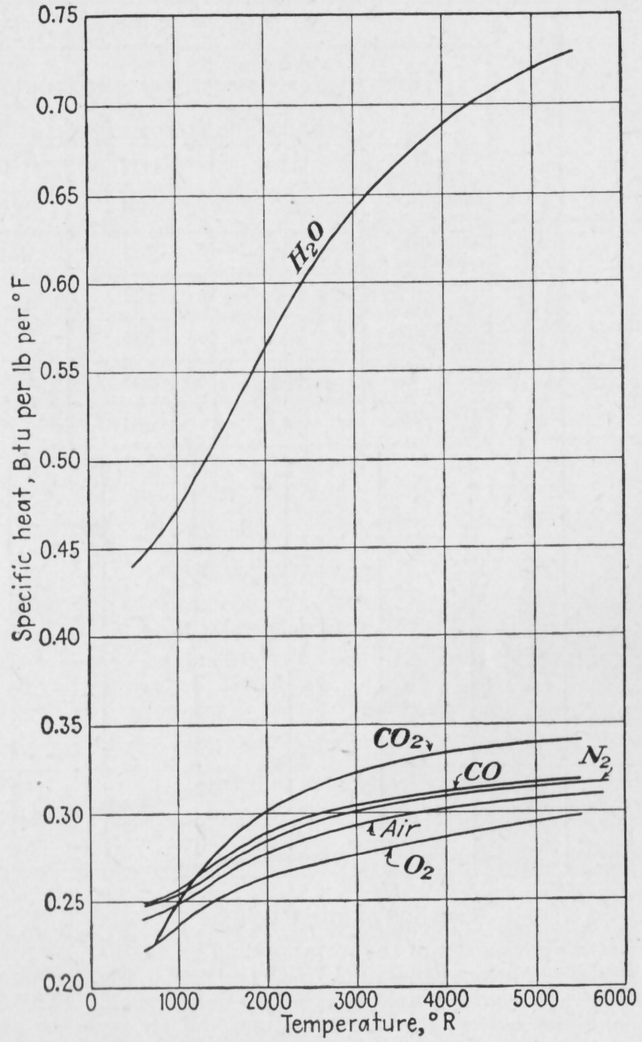

Fig. 34. Specific heat at constant pressure for each of the gases in the products of combustion. (*Plotted from data given by Heck, Mech. Eng., January,* 1940.)

the expansion stroke, this action follows much of the working portion of the cycle and hence does not help greatly in reducing the losses in power output and efficiency that result from dissociation.

Pressure-volume diagrams for the first, second, and fourth of the above successive approximations are shown in Fig. 35. The principal

factors involved in the calculations are given in a table in the upper right-hand corner of the diagram.

The Diesel Cycle. Rudolf Diesel, who devised the cycle and the engine type that bear his name, had visualized the cycle as an adiabatic

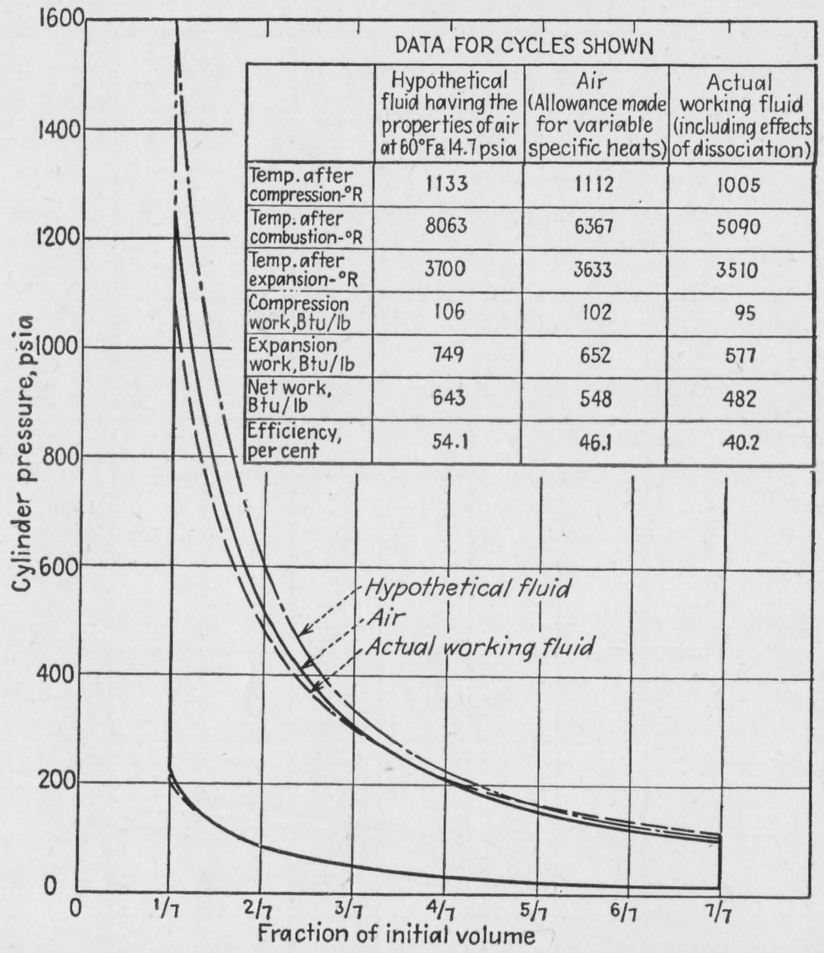

FIG. 35. Effect of the properties of the working fluid on the P-V diagram for an ideal Otto cycle with initial conditions equal to 14.7 psia and 520°R, a compression ratio of 7:1, and with 1190 Btu/lb of working fluid released by combustion. (This is the amount given by the chemically correct mixture of octane and air with the products in the gaseous state.)

compression of the air to the point where the high temperature developed would ignite fuel injected at that point. By controlling the fuel injection rate he had felt that the combustion rate could be so controlled as to maintain a constant pressure during combustion, as indicated in Fig. 36. In practice, it has been found that compression-ignition engines, particularly the higher speed models, operate best on

a close approximation to the constant-volume cycle because higher temperatures during combustion and hence better combustion conditions can be obtained in that way. As a result, most compression-ignition engines, while generally called "diesel" engines, actually approach a constant-volume cycle more closely than most spark-ignition engines.

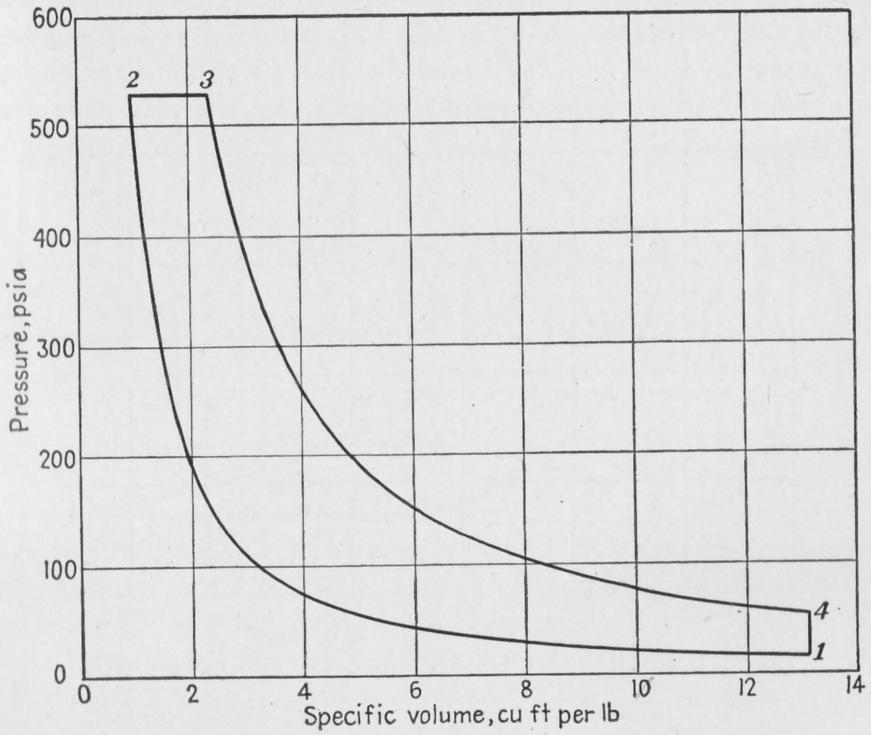

FIG. 36. Pressure-volume diagram for a classic diesel cycle.

The ideal efficiency of the classical diesel cycle can be derived by much the same procedure as that employed above for the Otto cycle. If heat is added from V_2 to V_3 so that the cutoff ratio may be defined as V_3/V_2 (see Fig. 36), such a derivation gives the expression

$$\text{Thermal eff} = \left[1 - \frac{1}{(V_1/V_2)^{k-1}}\right] \left\{\frac{(V_3/V_2)^k - 1}{k[(V_3/V_2) - 1]}\right\} \qquad (3)$$

This expression is the same as that for the constant-volume cycle except that the term within the brackets has been added. This term approaches unity as the cutoff ratio approaches 1. Thus the efficiency of the Diesel cycle approaches that of the constant-volume cycle as the amount of heat added approaches zero, *i.e.*, at low power outputs. As the amount of heat added during each cycle is increased,

the efficiency of the idealized cycle falls off rather rapidly. This indicates one reason for the general practice of operating diesel engines on a cycle much closer to the Otto than the diesel cycle.

Constant-pressure Cycle. A number of new types of prime mover operating on a constant-pressure cycle (also called the *Joule,* or *Brayton,* cycle) have come into use in recent years. These include the gas turbine and several types of thermal-jet-aircraft power plant. A diagram for this cycle is shown in Fig. 37. Air is compressed adiabatically in a compressor from 1 to 2 and passed through a combustion chamber where fuel is burned at constant pressure so that the volume increases

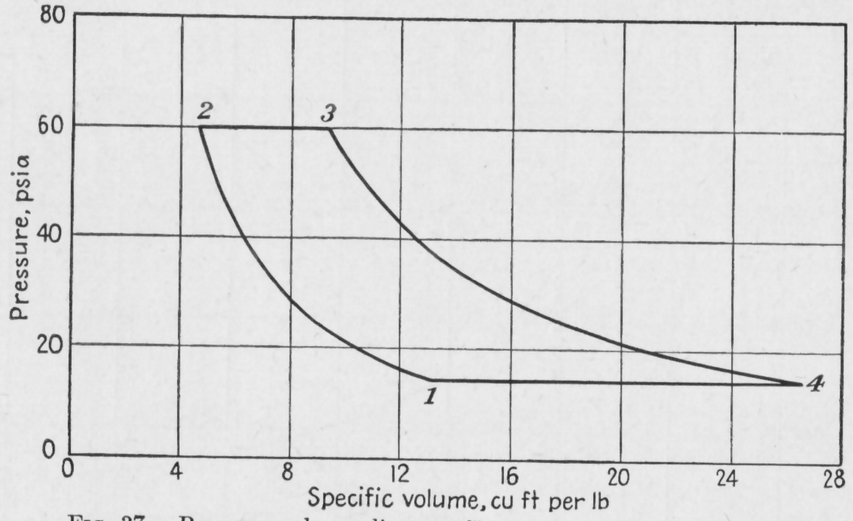

FIG. 37. Pressure-volume diagram for a constant-pressure cycle.

from 2 to 3, and the products of combustion are expanded adiabatically through a turbine from 3 to 4 and exhausted to atmospheric pressure. If the compression and expansion processes are isentropic, the characteristics of the cycle can be derived in much the same way as those for the constant-volume or Otto cycle.

From elementary thermodynamics, the work input to the gas flowing through the compressor is

$$\text{Work input} = Wc_p(T_2 - T_1)$$

The temperature rise in the combustion chamber may be related to the heat released during combustion by

$$\text{Heat released} = Q = Wc_p(T_3 - T_2)$$

The work output from the gas as it expands through the turbine is given by

$$\text{Work output} = Wc_p(T_3 - T_4)$$

The thermal efficiency of the cycle is then

$$\text{Thermal eff} = \frac{Wc_p(T_3 - T_4) - Wc_p(T_2 - T_1)}{Wc_p(T_3 - T_2)}$$

But

$$T_3 = T_4\left(\frac{P_3}{P_4}\right)^{\frac{k-1}{k}} \qquad \text{and} \qquad T_2 = T_1\left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}}$$

Hence

$$\text{Thermal eff} = \frac{[T_4(P_3/P_4)^{\frac{k-1}{k}} - T_4] - [T_1(P_2/P_1)^{\frac{k-1}{k}} - T_1]}{T_4(P_3/P_4)^{\frac{k-1}{k}} - T_1(P_2/P_1)^{\frac{k-1}{k}}}$$

But $P_3/P_4 = P_2/P_1$; hence the above expression can be reduced to

$$\text{Thermal eff} = \frac{T_4[(P_2/P_1)^{\frac{k-1}{k}} - 1] - T_1[(P_2/P_1)^{\frac{k-1}{k}} - 1]}{(T_4 - T_1)(P_2/P_1)^{\frac{k-1}{k}}}$$

$$= 1 - \frac{1}{(P_2/P_1)^{\frac{k-1}{k}}} \tag{4}$$

Factors Affecting the Efficiency of Ideal Cycles. The preceding derivations have shown that the thermal efficiencies of the three basic cycles employed in combustion engines depend on two factors, the compression ratio and the ratio of the specific heats. Since by far the most common working fluid consists of air or the products of combustion, the value of the ratio of the specific heats depends primarily on the relative proportions of the fuel and air. The curves in Fig. 38 show the importance of the effects of this factor on the efficiency of the constant-volume cycle with a working fluid consisting of air during the compression portion of the cycle and of the products of combustion of air and fuel during the expansion. Full allowances were made for dissociation and the variable specific heats. The loss in efficiency with an increase in the amount of fuel at any given compression ratio results partly from the fact that the specific heats of the combustion products at any given temperature increase with the percentage of water vapor and partly from the fact that, the greater the proportion of fuel burned, the higher the temperature after combustion. The latter acts to increase both the average specific heats and the amount of dissociation. If the percentage of fuel is increased above the chemically correct value, the above relations do not apply but the losses arising

from incomplete combustion have even greater effects. While the curves in Fig. 38 are for the constant-volume cycle, those for the constant-pressure cycle are almost identical while those for the classic deisel cycle are very similar.

The efficiency of the three basic cycles discussed in this chapter are compared in Fig. 39 for representative values of the fuel-air ratio,

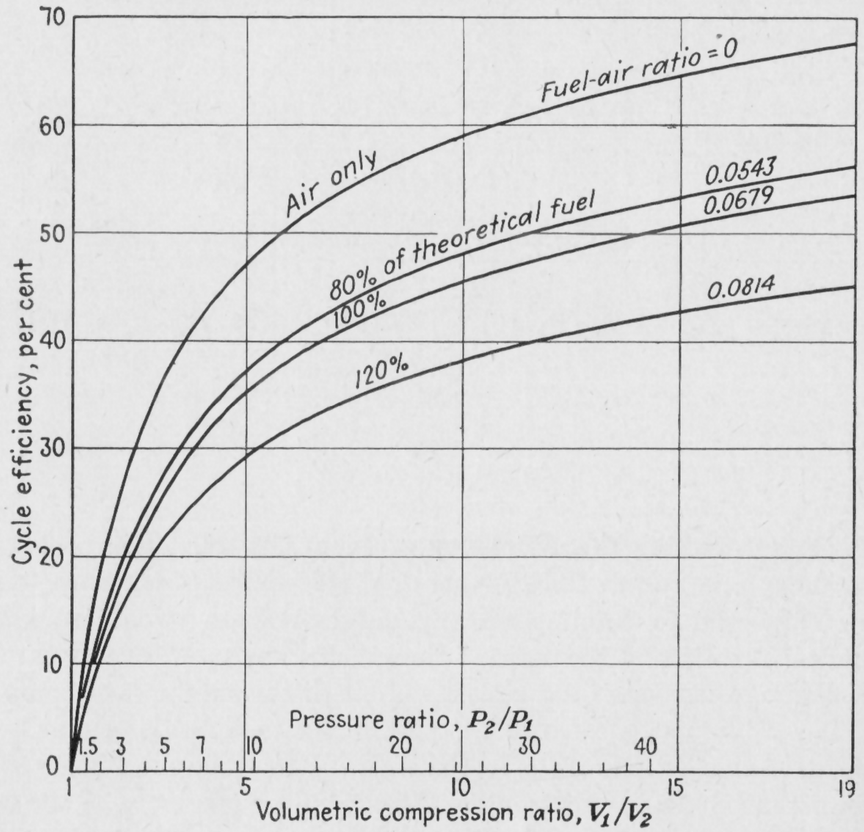

FIG. 38. Effects of compression ratio and fuel-air ratio on the thermal efficiency of the constant-volume cycle. Full allowances are made for variable specifics and dissociation in the products of combustion. The data are for a mixture initially consisting of octene vapor and air at 32°F and 14.7 psia.

the initial temperature, and the initial pressure. Reference to Eqs. (1) and (4) discloses that the efficiency of the constant-pressure cycle is the same as that of the constant-volume cycle at a given compression ratio because, for an isentropic compression,

$$\left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}} = \left(\frac{V_1}{V_2}\right)^{k-1}$$

The higher efficiency of the constant-pressure cycle in Fig. 39 does

not contradict the identity of Eqs. (1) and (4)—with the actual working fluid the higher temperature range of the constant-volume cycle increases both the mean effective specific heats and the losses caused by dissociation.

The reason why the efficiency of the diesel cycle is much lower than that of either of the other two in Fig. 39 can be visualized read-

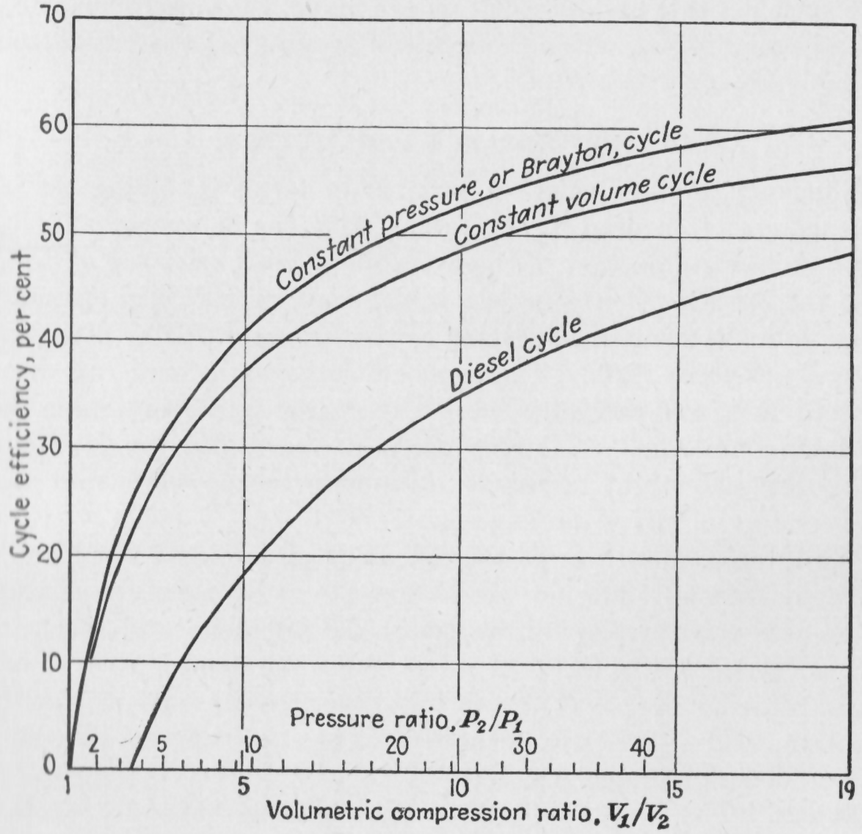

FIG. 39. Comparison of the thermal efficiencies of three ideal cycles for a fuel-air ratio of 0.0543. Full allowances are made for variable specific heats and dissociation in the products of combustion.

ily by comparing Figs. 33, 36, and 37. The indicator-diagram area for the diesel cycle is inherently smaller because area is lost at both ends of the diagram instead of at only one end as in the other two cycles. The efficiency of the diesel cycle drops at an especially rapid rate as the compression ratio is reduced below about 10:1 because the amount of isentropic expansion possible is reduced to such a small value.

Construction of Indicator Diagrams. If a P-V diagram is to be constructed for a combustion-engine cycle, the calculation of a large

number of intermediate points for the compression and expansion processes can be avoided by plotting the points computed for the four "corners" of the diagram on logarithmic coordinates. The points may then be connected by straight lines. Figure 67 shows a diagram of this sort. If a linear P-V diagram is desired, it can be replotted easily from the logarithmic diagram. Even if full allowances for variable specific heats and dissociation are made, the isentropics will still plot as straight lines within the limits of accuracy of any commercially available log-log graph paper.

PERFORMANCE PARAMETERS

Efficiency of the Cycle. A number of terms involving the word *efficiency* are often used in connection with engine cycles. Probably the three most important of these are the *thermal efficiency of the ideal cycle*, the *thermal efficiency of the actual cycle*, or *indicated thermal efficiency*, and the ratio of the latter to the former, a factor often called the *cycle-efficiency ratio*. The cycle-efficiency ratio may run over 90 per cent in a well-designed large-bore engine under favorable speed and load conditions. It is very useful as a criterion for determining the course that might be taken to improve the power output or the fuel economy of any given engine.

While such quantities as the efficiency of the ideal cycle and the indicated thermal efficiency are invaluable as a means for analyzing engine-performance possibilities, most development and service-test work is carried out in terms of performance parameters such as brake and indicated horsepower, mean effective pressure, specific fuel consumption, and volumetric efficiency. These parameters are defined in the following paragraphs.

Power. The power output of an engine is generally expressed either as *indicated horsepower*, or as *brake horsepower*. The former is the power developed in the cylinders, while the latter is the net power output to the drive shaft. Both terms are commonly abbreviated to ihp and bhp. The ihp could be computed from the net area of an indicator card, although that method is seldom employed in automotive-engine work because it is very difficult to take indicator cards. The usual procedure for determining indicated output is to make use of the difference between the ihp and the bhp, viz., the *friction horsepower* (fhp). This can be determined accurately enough for practical purposes by using a dynamometer to "motor" an engine after it has been well "warmed up." The bhp added to the fhp gives the ihp. (The term *motoring an engine with a dynamometer* is used when an electric dynamometer is operated as a motor to drive an engine without the cylinders firing.)

Mean Effective Pressure. The mep is that hypothetical pressure which, if it acted throughout the power stroke, would do the same amount of work as the cycle in question. That is, in four-cycle engines, the mep is given by the expression

$$\text{Mep} = \frac{\text{hp} \times 33{,}000}{\left(\frac{\text{rpm}}{2}\right)\left(\frac{\text{stroke}}{12}\right)(\text{piston area})(\text{number of cylinders})}$$

$$= \frac{\text{hp} \times 792{,}000}{(\text{rpm})(\text{piston displacement, in.}^3)} \tag{5}$$

Similarly, in two-cycle engines, the mep is given by:

$$\text{Mep} = \frac{\text{hp} \times 396{,}000}{(\text{rpm})(\text{piston displacement, in.}^3)}$$

It follows directly that the *indicated mean effective pressure*, or *imep*, can be computed very easily from dynamometer data for ihp without taking an indicator card and determining the average ordinate. While it is a somewhat fictitious quantity, the *brake mean effective pressure*, or *bmep*, is even more widely used. It is obtained simply by substituting values for bhp instead of ihp in Eq. 5.

Specific Fuel Consumption. The fuel-economy characteristics of an engine are generally expressed in terms of specific fuel consumption in pounds of fuel consumed per horsepower-hour. Again the expression is usually prefaced with "brake" or "indicated" depending on the basis for the value and is often abbreviated to bsfc or isfc.

Volumetric Efficiency. The maximum amount of air that an engine will handle is one of the most important factors limiting its power output. This follows from the fact that only a certain maximum amount of fuel can be burned effectively with a given quantity of air, and thus only a certain rate of heat release per pound of air can be obtained. While it is not difficult to increase the fuel flow rate to an engine, it is very difficult to increase the air flow beyond the amount that the engine naturally tends to induce. The product of the piston displacement and the rpm gives a first approximation to the air flow rate to the cylinders of a two-cycle engine. Half that quantity would be the ideal air flow rate to a four-cycle engine because each cylinder of the latter would experience a cycle only once every other revolution. The *volumetric efficiency* has been defined as the

ratio of the amount of air actually induced to this ideal quantity. That is, for four-cycle engines,

$$\text{Volumetric eff} = \frac{\text{actual engine air flow, cfm}}{\left(\frac{\text{rpm}}{2}\right)\left(\frac{\text{piston displacement, in.}^3}{1{,}728}\right)} \tag{6}$$

The volume of the air flowing into the engine in the above expression is generally determined on the basis of prevailing atmospheric conditions, although in some instances—notably for supercharged engines—it may be based on intake-manifold temperature and pressure.

Dilution of the fresh charge by the residual exhaust gas, heat transfer from the hot cylinder walls to the fresh charge during the intake stroke, and the pressure drop through the intake manifold and port all act to lower the volumetric efficiency to a value of the order of 75 to 90 per cent in a well-designed engine. These effects are discussed in Chap. XIII.

Fuel-Air Ratio. Figure 38 showed the great importance of the relative proportions of fuel and air in the working fluid of combustion engines from the standpoint of the cycle efficiency. Other vital factors, particularly the gas temperatures after combustion, are directly dependent on the same important quantity.

The relative proportions of fuel and air in the products of combustion may be expressed either as the ratio of the weight of fuel to that of the air, or vice versa. Steam-power-plant-furnace practice has been to use the air-fuel ratio. Most early automotive work followed the same convention. A number of definite advantages of the inverse term, the fuel-air ratio, led first to its widespread use in the aircraft field and, more recently, in the automotive field, where it is now the more generally used of the two. The newest combustion-engine field, that of gas turbines and jet engines, has used the fuel-air ratio almost exclusively.

There are several important advantages to the use of the fuel-air ratio instead of the air-fuel ratio. For one thing, in all types of engine the air flow under a particular set of operating conditions tends to remain constant, while the fuel flow is subject to variation. Thus the better parameter is the fuel-air ratio, for it varies directly rather than inversely with the prime operating variable. Again, an essentially straight line relationship results if any of several important parameters such as the heat released by combustion per pound of mixture are plotted against the fuel-air ratio for mixtures leaner than the chemically correct value (see Figs. 45 and 46). Further, an air-fuel-

ratio scale is highly nonlinear and tends greatly to expand the lean-mixture portion of the mixture range. Therefore, because it is the more widely used, because it is the more natural parameter, and to avoid confusion, the fuel-air rather than the air-fuel ratio will be used exclusively in this text.

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3. Goodenough, G. A., and J. B. Baker: A Thermodynamic Analysis of Internal Combustion Engine Cycles, *Univ. Ill. Eng. Expt. Sta. Bull.* 160, 1927.

Problems

1. Determine the pressure, temperature, and volume of the charge at the end of the compression stroke, the beginning of the expansion stroke, and the end of the expansion stroke for a constant-volume cycle in a cylinder having an initial volume of 1.0 ft.^{3} Take the initial pressure and temperature as being 14.7 psia and 520°R, the heat added during combustion as 1190 Btu/lb of working fluid, and the specific heats of the working fluid equal to those for air at standard conditions, viz., $c_p = 0.240$ Btu/(lb)(°F) and $c_v = 0.172$ Btu/(lb)(°F). Use a compression ratio of 6.0:1. Plot the resulting data first on log-log and then on linear coordinates.

2. Compute the work input during the compression stroke and the work output during the expansion stroke for the cycle in Prob. 1. Compare the computed results with values determined from corresponding areas on the linear P-V diagram.

3. Compute the thermal efficiency for the cycle in Prob. 1 from (*a*) the computed results in Prob. 2 and (*b*) from Eq. (1).

4. List three important thermodynamic differences between the actual working fluid in an internal-combustion engine and an idealized gas having the properties of air at room temperature.

5. Repeat the calculations of Prob. 1, using a mean effective specific heat for each of the three main processes, viz., compression, combustion, and expansion. Using the curve for air in Fig. 34, estimate the mean specific heat in each case for the temperature range indicated by the results of Prob. 1. Superimpose these results on the P-V diagrams in Prob. 1.

CHAPTER III

THERMODYNAMIC-CALCULATION METHODS FOR COMBUSTION-ENGINE CYCLES

As pointed out in the preceding chapter, the thermodynamic properties of the working fluid have a direct effect on the efficiency of the combustion-engine cycle. A number of different methods of carrying out calculations for idealized cycles were mentioned, and P-V diagrams for a set of typical cycles were shown in Fig. 35. The purpose of this chapter is to present a number of methods for carrying out these calculations as easily as possible with the achievement of good accuracy.

PROPERTIES OF THE WORKING FLUID

Composition. Since air constitutes the greater part of the working fluid in combustion engines, the composition of air as given in Table 2 is a matter of basic importance. For most purposes it is satisfactory to include the argon with the nitrogen and neglect the other components, only traces of which are present. The approximate composition of the air then becomes 21.0 per cent O_2 and 79.0 per cent N_2 by volume as indicated by the lower portion of the table.

The composition of the products of combustion is a complex function of fuel-air ratio, pressure, and temperature. The thermodynamic properties vary accordingly by amounts too great to be neglected if

TABLE 2. COMPOSITION OF AIR*

Molecule	Per cent by vol	Per cent by wt
Exact:		
Nitrogen	78.03	75.46
Oxygen	20.99	23.19
Argon	0.94	1.30
Carbon dioxide	0.03	0.04
Other gases	0.01	0.01
Approximate.		
Nitrogen	79.0	76.8
Oxygen	21.0	23.2

* "International Critical Tables," Vol. I, McGraw-Hill Book Company, Inc., New York, 1926.

more than a very rough approximation is desired. While most problems on engine cycles can be solved with charts, so that the composition of the combustion products need not be determined, it is important for the student to realize that the complexity of the relations governing composition is such that no simple general solution is possible and that either charts or some very involved procedure must be employed for any but the roughest sort of estimations.

TABLE 3. PROPERTIES OF COMBUSTION PRODUCTS*

Molecule	Chemical symbol	Molecular wt	Gas const R	
			Ft-lb/(lb)(°R)	Btu/(lb)(°R)
Nitrogen	N_2	28.016	55.16	0.07093
Argon	A	39.944	38.68	0.04974
Oxygen	O_2	32.000	48.32	0.06210
Water vapor	H_2O	18.016	85.83	0.11030
Carbon dioxide	CO_2	44.010	35.11	0.04510
Carbon monoxide	CO	28.010	55.18	0.07094
Hydrogen	H_2	2.016	766.6	0.9857
Methane	CH_4	16.042	96.33	0.1239
Values for air	....	28.95	53.34	0.06854

* "International Critical Tables," Vol. I, McGraw-Hill Book Company, Inc., New York, 1926.

Specific Heats. The specific heat at constant pressure was given in Fig. 34 as a function of temperature for the more common species of molecule in the products of combustion. Since the difference between the specific heat at constant pressure and that at constant volume is equal to the gas constant R and since the gas constant for each of these gases remains virtually unchanged over the temperature and pressure range encountered in combustion engines, no curve is given or required for the specific heat at constant volume. The gas constant R for any of the gases in the products of combustion can, of course, be determined by dividing the universal gas constant, 1,545, by the molecular weight of the gas.[1] Inasmuch as the value for R is generally given in foot-pounds per pound per degree Rankine, it should be divided by 778 to convert it to Btu per pound per degree Rankine if it is to be used to find the specific heat at constant volume in Btu per pound. Values for both the molecular weight and the gas constant are given in Table 3 for the principal gases present in the working fluid of a combustion engine.

Use of Mean Specific Heats. Rough approximations to the actual cycle can be made by assuming values for mean specific heats

of the working fluid. As in elementary thermodynamics, values for air at room temperature may be employed; $c_p = 0.24$ Btu/(lb)(°R), $c_v = 0.17$ Btu/(lb)(°R), $k = 1.4$, and $R = 53.3$. For the temperature range of a representative cycle, however, these values are low even if the working fluid were to consist only of air. The products of combustion further increase the specific heats so that a more representative set is $c_p = 0.300$ and $c_v = 0.231$, which give $c_p/c_v = k = 1.3$, and $R = (c_p - c_v)778 = 54.3$. While these constants are as representative as any, calculations made with them may yield temperatures and pressures differing by over 20 per cent from corresponding values calculated for the same cycle by a reasonably rigorous procedure.

Charts have been prepared to give values for the ratio of the mean specific heats so that this quantity can be determined accurately if the fuel-air ratio and the initial and final temperatures are given.[2,3] The temperature range can usually be approximated closely enough by a trial calculation. Calculations made with data of this sort are accurate except for the effects of dissociation, which are negligible if the temperatures do not exceed 2500°R.

Use of Enthalpy and Internal Energy. From the standpoint of this text, the accurate determination of the mean specific heats is just a means to an end, viz., the calculation of engine cycles. It happens that it is possible to make complete allowances for both variable specific heats and dissociation through the use of the quantities *internal energy* and *enthalpy* and at the same time greatly simplify the arithmetical work of cycle calculations. This can be seen by examining the derivations in the previous chapter. Only four types of thermodynamic process are of interest, viz., constant-volume heating, constant-pressure heating, isentropic compression or expansion without flow of the working fluid, and isentropic compression or expansion with flow of the working fluid. It further happens that the change in the internal energy is an exact measure of either the heat added during a constant-volume heating process or the work done during an isentropic compression (or expansion) without flow of the working fluid through the machine. Similarly, the change in enthalpy is an exact measure of either the heat added during a constant-pressure heating or the work done during an isentropic compression (or expansion) with flow of the working fluid through the machine during the process.

While all the above relations are demonstrated rigorously in elementary thermodynamics,[1] there is often some confusion as to the distinction between the use of internal energy and enthalpy. The difference between the two can be shown very clearly by considering

the isentropic compression of air in the cylinder of an air compressor. A P-V diagram for such a process is given in Fig. 40. The algebra can be simplified by considering the pressure on the crank side of the piston to be 0.0 psia. The work energy in Btu required for compression of the air from a to b without flow into or from the cylinder is represented by the area $abfg$, which from basic thermodynamics is

$$\text{Work of compression} = \frac{P_2V_2 - P_1V_1}{778(k-1)} \tag{7}$$

The work required to force the air to flow out of the cylinder at constant pressure (assuming a zero clearance volume) is represented

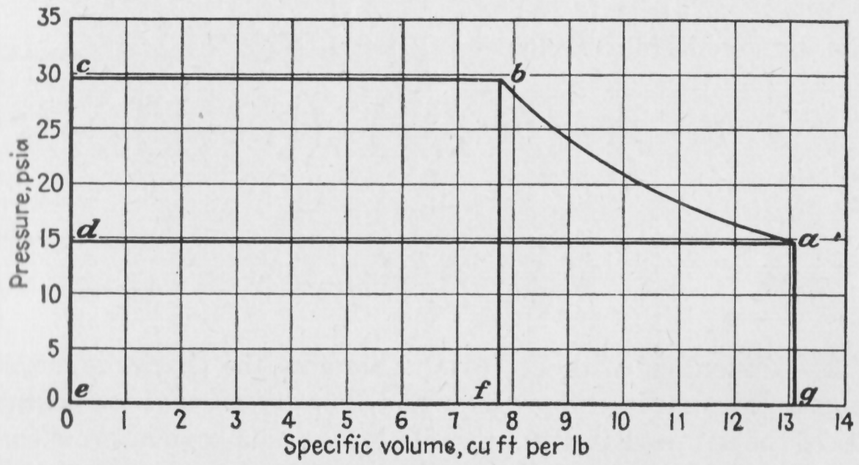


Fig. 40. Pressure-volume diagram for a piston-type air compressor with zero clearance volume.

by the area $bcef$ and is given by the product P_2V_2. As the piston moves back to take in a fresh charge of air, the air flowing into the cylinder does work on the piston, the amount in this instance being represented by the area $dage$, which is equal to the product P_1V_1. The net work required for the complete cycle of intake, compression, and delivery thus is represented by the area $abcd$. Algebraically, this is

$$\begin{aligned}\text{Net work for compression and delivery} &= \frac{P_2V_2 - P_1V_1}{778(k-1)} + \frac{P_2V_2}{778} - \frac{P_1V_1}{778} \\ &= \frac{P_2V_2 - P_1V_1}{778(k-1)} + \frac{k(P_2V_2 - P_1V_1) - (P_2V_2 - P_1V_1)}{778(k-1)} \\ &= \frac{k}{k-1}\frac{P_2V_2 - P_1V_1}{778}\end{aligned} \tag{8}$$

Equation (8) is the same as Eq. (7) except that it has been multiplied by the constant k.

The significance of the above from the standpoint of the terms internal energy and enthalpy (expressed in Btu) follows: By definition, the change in the internal energy U is

$$U_2 - U_1 = Wc_v(T_2 - T_1)$$

But one of the identities demonstrated in elementary thermodynamics is

$$Wc_v(T_2 - T_1) = \frac{P_2V_2 - P_1V_1}{778(k - 1)}$$

Again, by definition, the change in the enthalpy H is

$$\begin{aligned} H_2 - H_1 &= \left(U_2 + \frac{P_2V_2}{778}\right) - \left(U_1 + \frac{P_1V_1}{778}\right) \\ &= \frac{P_2V_2 - P_1V_1}{778(k - 1)} + \frac{P_2V_2}{778} - \frac{P_1V_1}{778} \\ &= \left(\frac{k}{k - 1}\right)\frac{P_2V_2 - P_1V_1}{778} \end{aligned}$$

This is identical with Eq. (8). Therefore the change in enthalpy during an isentropic compression and delivery process is identically equal to the net work in Btu. Similarly, the change in internal energy is equal to the work in Btu for an isentropic compression *without flow of the working fluid.* From the conservation-of-energy law, it follows that these relations hold for either an isentropic compression or an isentropic expansion in any type of machine. Thus it follows that it is not necessary to evaluate either the specific heats or their ratio k for each phase of the cycle if values for the internal energy and enthalpy are available from a chart or table. Methods of making cycle calculations using enthalpy and internal energy are presented in the following sections.

Deviations in the behavior of actual gases from the perfect-gas laws make it difficult to evaluate internal energy and enthalpy at temperatures near absolute zero. Since the lower temperature range is of little practical interest, a value of zero is usually assigned to the enthalpy at some convenient base temperature. The tables and charts given in this book use 400°R. Inasmuch as cycle-analysis problems are always concerned with the *change* in enthalpy or internal energy, the base temperature does not influence calculated results.

TABLES FOR THE THERMODYNAMIC PROPERTIES OF AIR

It is sometimes of interest to determine the characteristics of a cycle employing air only as a working fluid and to make proper allowance for its variable specific heats. This is particularly true in gas turbines and also in diesel engines at light loads. In both cases the fuel-air ratio is so low that the actual working fluid differs little from air. Calculations of this type can be made easily through the use of a set of air tables prepared by Keenan and Kaye.[4] Data have been abstracted from these air tables and presented in Table 4 to show their application. While the values in Table 4 are given only at 10° or 100° intervals, the original tables give data at 1° or 10° intervals for a greater range of temperatures. However, the data in Table 4 will serve for instructional purposes.

Basis for Air Tables. In the temperature range from 0° to 2500°F and for the corresponding pressure range of interest in combustion-engine work, air follows the perfect-gas laws very closely at any given temperature. This fact makes it possible to present the thermodynamic properties in a much simpler form than is possible for a substance, such as steam, that is used at temperatures in the vicinity of its boiling point where the gas constant varies so with the pressure. The basis for the Keenan and Kaye air tables is that all isentropics for a perfect gas are directly and simply related, *i.e.*, at any given temperature and volume the pressure is directly proportional to the weight of gas present. Therefore—irrespective of the density of the gas—its specific heats, internal energy, and enthalpy in Btu per pound are dependent only on the temperature. Further, the ratio of the initial to the final volume of a perfect gas if compressed isentropically from one given temperature to another is independent of its density. Similarly, the ratio of the initial to the final pressure for the same process is also independent of density. Therefore the pressures and volumes for all isentropics may be related by a set of values giving pressures and volumes relative to some datum condition.

This concept of *relative volumes* and *relative pressures* for isentropic compressions or expansions may be clarified by considering the relations between the volumes V_0, V_1, and V_2 for a given quantity of gas that is to be compressed isentropically from the datum temperature T_0 first to the temperature T_1 and then to the temperature T_2. From elementary thermodynamics we may define the quantity *relative volume*, V_r, by writing

$$\frac{T_0}{T_1} = \left(\frac{V_1}{V_0}\right)^{k-1} = (V_{r_1})^{k-1}$$

and

$$\frac{T_0}{T_2} = \left(\frac{V_2}{V_0}\right)^{k-1} = (V_{r_2})^{k-1}$$

where V_{r_1} and V_{r_2} are, respectively, the volumes at points 1 and 2 relative to V_0, the datum condition. If the temperature and volume at point 1 are known and the volume at point 2 is defined by the compression ratio as would be the case for a gasoline-engine cycle, for example, the temperature at point 2 can be easily determined if a table or chart is available in which the relative volume is given in terms of the temperature. The value for V_r at point 1 can be picked from the table at T_1. The relative volume at point 2 is then determined by dividing the value for V_{r_1} by the compression ratio, and the final temperature T_2 can be picked from the table once V_{r_2} is known. The validity of this procedure is easily demonstrated from the equations above. That is,

$$\frac{(V_1/V_0)^{k-1}}{(V_2/V_0)^{k-1}} = \frac{(V_{r_1})^{k-1}}{(V_{r_2})^{k-1}}$$

Simplifying,

$$\frac{V_1}{V_2} = \frac{V_{r_1}}{V_{r_2}}$$

Note that the datum temperature and volume cancel out—they are necessary only to provide a "starting point" for the table or chart in which the relative-volume data are given.

In the same way the pressure relative to that at the datum point can be determined and tabulated or plotted in terms of temperature. Values for the internal energy and enthalpy are also unique functions of temperature because, in all but the highest part of the range covered by Table 4, the amount of dissociation is negligible.

Use of Air Tables. The application of the air tables can be demonstrated readily by means of an example. Consider a constant-volume cycle as carried out in a reciprocating engine having a compression ratio of 13.9:1. The pressure and temperature at the beginning of the compression stroke may be taken as being 12.4 psia and 650°R, respectively, the fuel-air ratio as 0.010, and the lower heating value of the fuel as 19,190 Btu/lb. The calculations can be carried out most easily if set up in tabular form as in Table 5 for 1 lb of air. The given temperature and pressure can be recorded in the appropriate spaces and the initial volume computed from them. Since the quantity of air used is 1 lb, this volume is also the specific volume, a fact that will

TABLE 4. THERMODYNAMIC PROPERTIES OF AIR*

T °R	h	P_r	u	V_r
400	0	1.0000	−27.42	10,000
410	2.40	1.0902	−25.71	9,402
420	4.79	1.1860	−24.00	8,854
430	7.19	1.2877	−22.29	8,348
440	9.59	1.3954	−20.57	7,883
450	11.98	1.5095	−18.86	7,453
460	14.38	1.6301	−17.15	7,055
470	16.78	1.7575	−15.44	6,686
480	19.18	1.8919	−13.73	6,343
490	21.57	2.0334	−12.01	6,024
500	23.97	2.182	−10.30	5,727
510	26.37	2.339	− 8.58	5,451
520	28.77	2.504	− 6.87	5,192
530	31.17	2.676	− 5.15	4,951
540	33.57	2.857	− 3.44	4,725
550	35.98	3.047	− 1.72	4,512
560	38.38	3.246	− 0.01	4,313
570	40.78	3.454	1.71	4,126
580	43.19	3.671	3.43	3,950
590	45.59	3.898	5.15	3,784
600	48.00	4.135	6.87	3,628
610	50.41	4.382	8.59	3,481
620	52.81	4.639	10.32	3,341
630	55.22	4.907	12.04	3,209
640	57.63	5.187	13.76	3,085
650	60.04	5.478	15.49	2,967
660	62.46	5.780	17.22	2,855
670	64.87	6.094	18.95	2,749
680	67.28	6.421	20.67	2,648
690	69.70	6.760	22.40	2,552
700	72.11	7.112	24.13	2,461
710	74.53	7.477	25.87	2,374
720	76.95	7.855	27.60	2,291
730	79.38	8.248	29.34	2,213
740	81.80	8.654	31.08	2,138
750	84.23	9.075	32.82	2,066
760	86.65	9.511	34.56	1,997.7
770	89.08	9.962	36.30	1,932.3
780	91.51	10.428	38.05	1,869.9
790	93.95	10.911	39.80	1,810.1

TABLE 4. THERMODYNAMIC PROPERTIES OF AIR.*—(*Continued*)

T	h	P_r	u	V_r
800	96.38	11.410	41.55	1,752.9
810	98.82	11.925	43.30	1,698.2
820	101.26	12.457	45.05	1,645.7
830	103.70	13.006	46.81	1,595.4
840	106.14	13.574	48.57	1,547.1
850	108.59	14.160	50.33	1 500.7
860	111.04	14.764	52.09	1,456.3
870	113.49	15.387	53.86	1,413.6
880	115.94	16.029	55.62	1,372.5
890	118.40	16.692	57.40	1,333.0
900	120.86	17.374	59.17	1,295.0
910	123.32	18.077	60.94	1,258.5
920	125.78	18.802	62.72	1,223.3
930	128.25	19.548	64.50	1,189.4
940	130.72	20.316	66.29	1,156.7
950	133.19	21.106	68.07	1,125.3
960	135.67	21.919	69.87	1,094.9
970	138.14	22.756	71.66	1,065.7
980	140.63	23.617	73.45	1,037.4
990	143.11	24.502	75.25	1,010.1
1000	145.59	25.41	77.05	983.8
1010	148.09	26.35	78.86	958.3
1020	150.58	27.31	80.66	933.7
1030	153.07	28.30	82.47	910.0
1040	155.57	29.31	84.29	887.0
1050	158.07	30.35	86.10	864.8
1060	160.58	31.42	87.92	843.4
1070	163.09	32.52	89.74	822.6
1080	165.60	33.64	91.57	802.5
1090	168.11	34.80	93.40	783.0
1100	170.62	35.99	95.23	764.1
1110	173.15	37.21	97.06	745.8
1120	175.67	38.45	98.90	728.1
1130	178.20	39.74	100.74	710.9
1140	180.73	41.05	102.59	694.3
1150	183.26	42.40	104.43	678.1
1160	185.79	43.78	106.28	662.5
1170	188.33	45.19	108.14	647.3
1180	190.87	46.64	109.99	632.5
1190	193.42	48.12	111.85	618.2

TABLE 4. THERMODYNAMIC PROPERTIES OF AIR.*—(*Continued*)

T	h	P_r	u	V_r
1200	195.96	49.64	113.71	604.3
1210	198.52	51.20	115.58	590.8
1220	201.07	52.79	117.45	577.7
1230	203.63	54.43	119.32	565.0
1240	206.19	56.10	121.20	552.6
1250	208.75	57.81	123.07	540.6
1260	211.32	59.56	124.96	528.9
1270	213.89	61.35	126.84	517.5
1280	216.47	63.19	128.73	506.4
1290	219.04	65.07	130.62	495.7
1300	221.62	66.98	132.52	485.2
1310	224.21	68.95	134.42	475.0
1320	226.79	70.95	136.32	465.1
1330	229.38	73.01	138.22	455.4
1340	231.98	75.10	140.13	446.0
1350	234.57	77.25	142.04	436.9
1360	237.17	79.44	143.95	428.0
1370	239.78	81.68	145.87	419.3
1380	242.38	83.97	147.79	410.8
1390	244.99	86.32	149.71	402.6
1400	247.60	88.71	151.64	394.6
1410	250.22	91.15	153.57	386.7
1420	252.84	93.64	155.51	379.1
1430	255.46	96.19	157.44	371.7
1440	258.08	98.79	159.38	364.4
1450	260.71	101.45	161.32	357.3
1460	263.34	104.16	163.27	350.4
1470	265.98	106.93	165.22	343.7
1480	268.61	109.75	167.17	337.1
1490	271.25	112.64	169.12	330.7
1500	273.89	115.58	171.08	324.5
1510	276.54	118.58	173.04	318.3
1520	279.19	121.64	175.00	312.4
1530	281.84	124.77	176.97	306.6
1540	284.50	127.96	178.94	300.9
1550	287.15	131.21	180.91	295.3
1560	289.81	134.52	182.88	289.9
1570	292.47	137.91	184.86	284.6
1580	295.14	141.36	186.84	279.4
1590	297.81	144.87	188.83	274.4

TABLE 4. THERMODYNAMIC PROPERTIES OF AIR.*—(*Continued*)

T	h	P_r	u	V_r
1600	300.48	148.46	190.81	269.4
1610	303.16	152.11	192.80	264.6
1620	305.84	155.83	194.80	259.9
1630	308.52	159.63	196.79	255.3
1640	311.20	163.50	198.79	250.8
1650	313.88	167.44	200.79	246.4
1660	316.58	171.45	202.79	242.0
1670	319.27	175.55	204.80	237.8
1680	321.96	179.72	206.81	233.7
1690	324.66	183.96	208.82	229.7
1700	327.36	188.29	210.84	225.7
1710	330.06	192.69	212.85	221.9
1720	332.77	197.18	214.87	218.1
1730	335.47	201.74	216.89	214.4
1740	338.19	206.40	218.92	210.8
1750	340.90	211.13	220.95	207.2
1760	343.61	215.95	222.97	203.7
1770	346.33	220.86	225.01	200.4
1780	349.05	225.86	227.04	197.0
1790	351.78	230.94	229.09	193.8
1800	354.50	236.1	231.12	190.59
1810	357.23	241.4	233.17	187.45
1820	359.96	246.7	235.21	184.40
1830	362.70	252.2	237.26	181.41
1840	365.43	257.7	239.31	178.47
1850	368.17	263.4	241.36	175.58
1860	370.92	269.1	243.43	172.78
1870	373.66	275.0	245.48	170.01
1880	376.41	280.9	247.55	167.32
1890	379.16	286.9	249.61	164.67
1900	381.90	293.1	251.68	162.07
1910	384.66	299.3	253.74	159.51
1920	387.42	305.7	255.81	157.03
1930	390.18	312.1	257.89	154.58
1940	392.94	318.7	259.96	152.18
1950	395.70	325.4	262.04	149.82
1960	398.47	332.2	264.12	147.52
1970	401.23	339.1	266.20	145.24
1980	404.00	346.1	268.28	143.03
1990	406.78	353.2	270.38	140.85

TABLE 4. THERMODYNAMIC PROPERTIES OF AIR.*—(*Continued*)

T	h	P_r	u	V_r
2000	409.55	360.4	272.46	138.72
2010	412.33	367.8	274.56	136.62
2020	415.11	375.3	276.65	134.57
2030	417.89	382.9	278.74	132.55
2040	420.67	390.6	280.84	130.58
2050	423.46	398.4	282.94	128.63
2060	426.25	406.4	285.05	126.72
2070	429.04	414.5	287.15	124.86
2080	431.83	422.7	289.26	123.01
2090	434.62	431.0	291.36	121.22
2100	437.42	439.5	293.48	119.45
2110	440.22	448.1	295.59	117.71
2120	443.02	456.9	297.71	116.01
2130	445.82	465.7	299.82	114.33
2140	448.62	474.7	301.94	112.69
2150	451.42	483.9	304.06	111.08
2160	454.23	493.2	306.18	109.50
2170	457.04	502.6	308.30	107.94
2180	459.86	512.2	310.43	106.41
2190	462.67	521.9	312.56	104.91
2200	465.48	531.7	314.68	103.43
2210	468.30	541.7	316.82	101.98
2220	471.12	551.9	318.95	100.56
2230	473.94	562.2	321.09	99.16
2240	476.76	572.6	323.22	97.79
2250	479.59	583.3	325.37	96 44
2260	482.41	594.0	327.50	95.11
2270	485.24	604.9	329.65	93.81
2280	488.07	616.0	331.79	92.53
2290	490.90	627.3	333.93	91.27
2300	493.73	638.7	336.08	90.03
2310	496.57	650.2	338.23	88.81
2320	499.41	662.0	340.39	87.62
2330	502.24	673.8	342.53	86.44
2340	505.08	685.9	344.69	85.29
2350	507.92	698.1	346.84	84.15
2360	510.77	710.5	349.01	83.03
2370	513.61	723.1	351.16	81.94
2380	516.46	735.9	353.32	80.85
2390	519.31	748.8	355.49	79.79

TABLE 4. THERMODYNAMIC PROPERTIES OF AIR.*—(*Continued*)

T	h	P_r	u	V_r
2400	522.16	761.9	357.65	78.75
2500	550.74	903.3	379.38	69.19
2600	579.48	1064.8	401.26	61.05
2700	608.36	1248.3	423.29	54.07
2800	637.37	1456.0	445.45	48.08
2900	666.51	1690.3	467.73	42.89
3000	695.77	1953	490.14	38.39
3100	725.14	2248	512.66	34.47
3200	754.62	2577	535.27	31.04
3300	784.19	2943	557.99	28.03
3400	813.85	3349	580.80	25.38
3500	843.61	3798	603.71	23.04
3600	873.45	4293	626.70	20.96
3700	903.38	4839	649.76	19.117
3800	933.38	5438	672.91	17.468
3900	963.46	6095	696.14	15.998
4000	993.61	6813	719.43	14.678
4100	1023.82	7596	742.80	13.494
4200	1054.11	8450	766.23	12.427
4300	1084.46	9377	789.72	11.464
4400	1114.87	10384	813.28	10.593
4500	1145.34	11475	836.89	9.804
4600	1175.86	12655	860.56	9.087
4700	1206.44	13929	884.28	8.436
4800	1237.07	15303	908.06	7.841
4900	1267.75	16783	931.88	7.299
5000	1298.48	18373	955.75	6.803
5100	1329.25	20080	979.68	6.349
5200	1360.07	21920	1003.65	5.932
5300	1390.94	23880	1027.66	5.549
5400	1421.85	25980	1051.72	5.197
5500	1452.80	28220	1075.81	4.872
5600	1483.80	30620	1099.95	4.572
5700	1514.83	33170	1124.13	4.296
5800	1545.91	35890	1148.35	4.040
5900	1577.02	38790	1172.61	3.803
6000	1608.16	41870	1196.90	3.583
6100	1639.35	45140	1221.23	3.379
6200	1670.57	48610	1245.60	3.189
6300	1701.83	52290	1270.00	3.012
6400	1733.12	56180	1294.44	2.848
6500	1764.45	60310	3118.91	2.694

* KEENAN, J. H., and J. KAYE, "Thermodynamic Properties of Air," 1st ed., John Wiley & Sons, Inc., New York, 1945.

TABLE 5. TABULATED CALCULATIONS FOR A CONSTANT-VOLUME AIR CYCLE

Given: Initial pressure = 12.4 psia; initial temperature = 650°R; compression ratio = 13.9:1; fuel-air ratio = 0.01; lower heating value of fuel = 19,190 Btu/lb.

Condition of charge in cylinder	(1) Beginning compression stroke	(2) End compression stroke	(3) Beginning expansion stroke	(4) End expansion stroke
Temperature, °R	650	1,732	2,635	1,075
Pressure, psia	12.4	459	699	20.55
Specific volume, ft³/lb	19.4	1.398	1.398	19.4
Relative pressure,	5.478	202.67	1,126.4	33.08
Relative volume,	2,967	213.5	58.48	813.5
Internal energy, Btu/lb	15.49	217.3	409.0	90.65

Energy input to gas on compression stroke = 217.3 − 15.49 = 201.8 Btu/lb
Energy input to piston on expansion stroke = 409.0 − 90.65 = 318.4 Btu/lb
Net input to piston = net E = 318.4 − 201.8 = 116.6 Btu/lb

$$\text{Thermal eff} = \frac{116.6}{191.9} = 60.7 \text{ per cent}$$

$$\text{Imep} = \frac{778 \times 116.6}{144(19.4 - 1.4)} = 35 \text{ psi}$$

$$\text{Isfc} = \frac{0.010 \times 2{,}545}{116.6} = 0.218 \text{ lb/hp-hr}$$

prove useful later. The relative pressure, relative volume, and internal energy can be read from the air tables. The conditions at the end of the isentropic compression can be found by dividing both the specific volume and the relative volume by 13.9 and recording the new values. The temperature after compression can be determined by finding the new relative volume in the air tables. The internal energy and relative pressure should also be read from the table and recorded. The value of the pressure can then be computed easily because the pressures on an isentropic are directly proportional to the relative pressures in the table. Since the data are for 1 lb of air, the increase in internal energy brought about by combustion then becomes the product of the fuel-air ratio and the heating value of the fuel, viz., 0.010 × 19,190 = 191.9 Btu. This increase in internal energy when added to the internal energy at the end of the compression stroke gives the internal energy at the beginning of the expansion stroke. The temperature together with the relative pressure and volume for the point on the new isentropic can also be recorded. The calculations for the expansion can be carried out in the same way as those for the compression to complete the data for the conditions at each of the four principal points in the cycle.

The energy input to the charge from the piston during the compression stroke can be determined by taking the difference between the values for the internal energy at the beginning and the end of the stroke. Similarly, the energy output from the charge to the piston can be determined for the expansion stroke. The difference between these energy exchanges for the expansion and compression strokes is then the net energy delivered to the piston, a quantity referred to in Table 5 as the *net E*. The efficiency of the cycle may be found by dividing this net energy by the energy of combustion, *i.e.*, the heat-energy input to the cycle.

All the above energies are in Btu per pound of air. The actual weight of air in a particular cylinder may be found from the cylinder volume and the specific volume of the air. The energy output per cycle can be determined for any particular cylinder size from the actual weight of air in the cylinder and the net energy output, or net E, per pound of air.

Calculation of IMEP and ISFC. The indicated mean effective pressure (imep) can be readily determined because, by definition, the product of the imep, the piston area, and the stroke is equal to the output of the cycle, *i.e.*:

$$\text{Imep} \times 144\ (\text{Area, ft}^2) \times \text{stroke, ft} = 778\ (\text{net E, Btu/lb air}) \times \text{lb air in cylinder}$$

But, where v_1 and v_2 are the specific volumes given by the chart for the beginning and the end of the compression stroke:

$$\frac{(\text{Area})(\text{stroke})}{\text{Lb air in cylinder}} = v_1 - v_2$$

From this it follows that

$$\text{Imep} = \frac{778\ (\text{net } E, \text{ Btu/lb air})}{144(v_1 - v_2)}$$

The indicated specific fuel consumption (isfc) may be determined since, by definition,

$$\text{Isfc} = \frac{\text{lb fuel/hr}}{\text{ihp}} = \frac{(F/A)(\text{lb air/hr})}{\dfrac{(\text{net } E, \text{ Btu/lb air})(\text{lb air/hr})}{2{,}545}}$$

The engine air-consumption rate cancels out; hence

$$\text{Isfc} = \frac{(F/A) \times 2{,}545}{\text{net } E, \text{ Btu/lb air}}$$

The calculations for these quantities are included at the bottom of Table 5.

THERMODYNAMIC CHARTS FOR THE PRODUCTS OF COMBUSTION

While it is possible to carry out calculations for an idealized cycle from basic data and make proper allowances for dissociation and the variable specific heats of the different species of molecule in the actual working fluid, such calculations are almost unbelievably tedious and time consuming. Hottel and his coworkers have prepared thermodynamic charts for the products of combustion of air and a representative hydrocarbon to include the effects of dissociation and variable specific heats and thus tremendously simplify accurate calculation work.[5,6] These charts look something like a Mollier diagram for the thermodynamic properties of steam. One of these Hottel charts is presented in Fig. 41, while a set of five charts made to a large scale to facilitate reading are folded in a pocket in the back of this book. The application of charts of this type is somewhat restricted by the fact that, for reciprocating-engine work, charts are available for only a limited number of particular fuel-air ratios, although a somewhat similar chart good for any fuel-air ratio has been prepared for gas-turbine work for the temperature range from 360 to 2520°R, where the amount of dissociation is negligible.[7] While the ratio of hydrogen to carbon in the fuel is another variable whose influence cannot be taken into consideration in the high-temperature range with this type of chart, its effect has been shown to be quite small.[8] Since the type of chart in Fig. 41 is a very powerful tool suited to a wide variety of calculations, the bases for its use are given in the following section.

Bases for Charts. The chart in Fig. 41 and those at the back of the book were prepared for the products of combustion of air and octene, a hydrocarbon representative of those in gasoline. A chart was constructed for each of five relative proportions of fuel and air, viz., mixtures containing 80, 90, 100, 110, and 120 per cent of the chemically correct amount of fuel. In all five charts the basic quantity of working fluid is that given by 1 lb of air in the fuel-air mixture before combustion. Taking the chart for the chemically correct mixture as an example, the amount of fuel present in the original mixture before combustion thus is 0.0677 lb, while the total weight of the products of combustion is 1.0677 lb. Values for these quantities are presented with other pertinent data in Table 6 for each of the five charts at the back of the book.

The coordinate system for the charts is internal energy vs. entropy.

E = INTERNAL ENERGY, B.T.U. ABOVE CO_2, H_2O (VAPOR), O_2, AND AIR-N_2 AT 100°F

Q_r = INT ENERGY OF COMBUSTION, AT 100°F., OF UNBURNED FUEL IN THE EQUILIBRIUM MIXTURE AT T. WHEN T = 2880°R., Q_r = 1, 2, 5, AT S = 0.4, 0.6, 0.8, RESPECTIVELY

$E_s = E - Q_r$

H = ENTHALPY, $E + J(PV)$

$H_s = E_s + J(PV)$ [$\therefore H = H_s + Q_r$]

P = PRESSURE, LBS./SQ. IN (DASHED LINES)

V = VOLUME, CU. FT (SOLID LINES)

S = ENTROPY, ABOVE CO_2, H_2O (VAPOR) O_2, AND AIR-N_2, EACH AT ONE ATMOSPHERE, 100°F

T = TEMPERATURE, °R = (°F + 460)

FUEL = $(CH_2)_x$

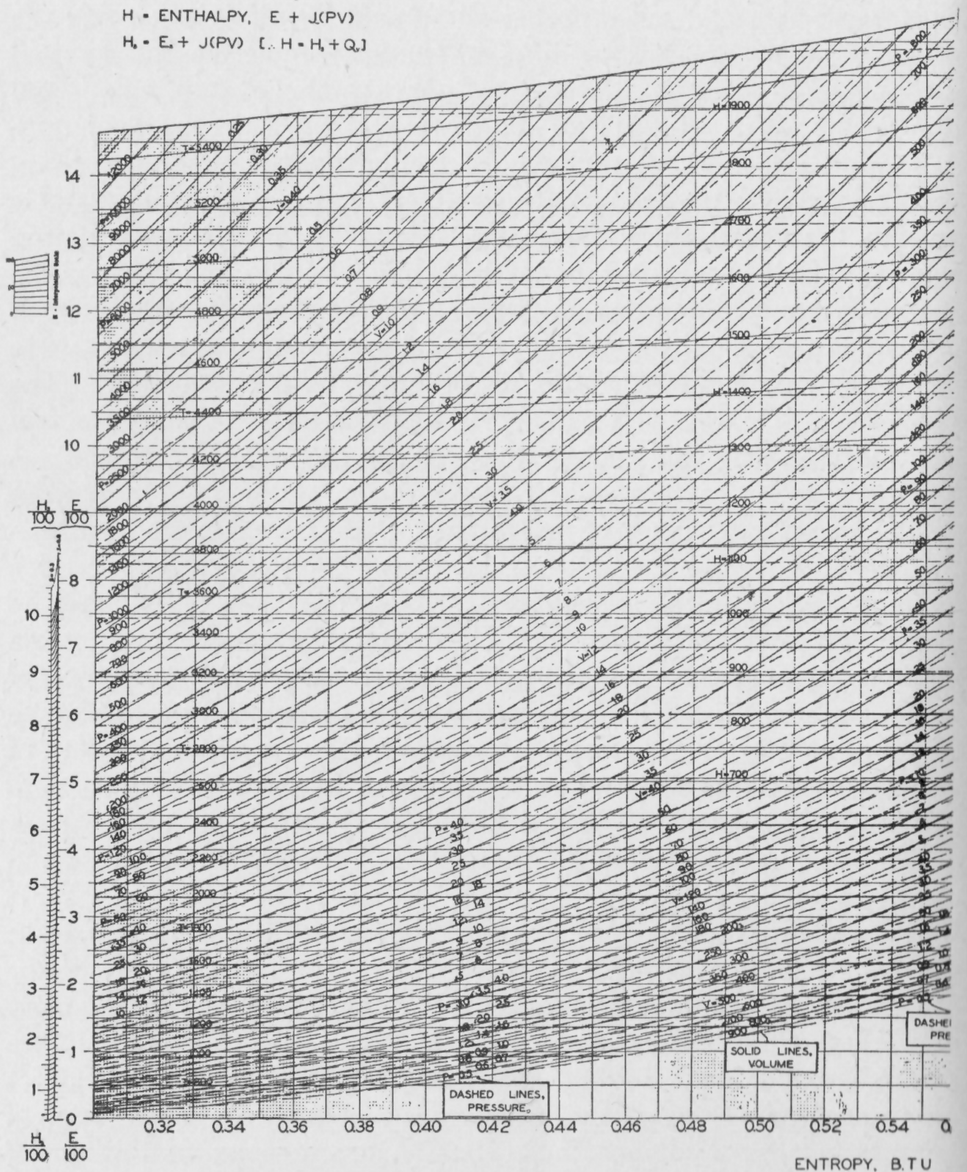
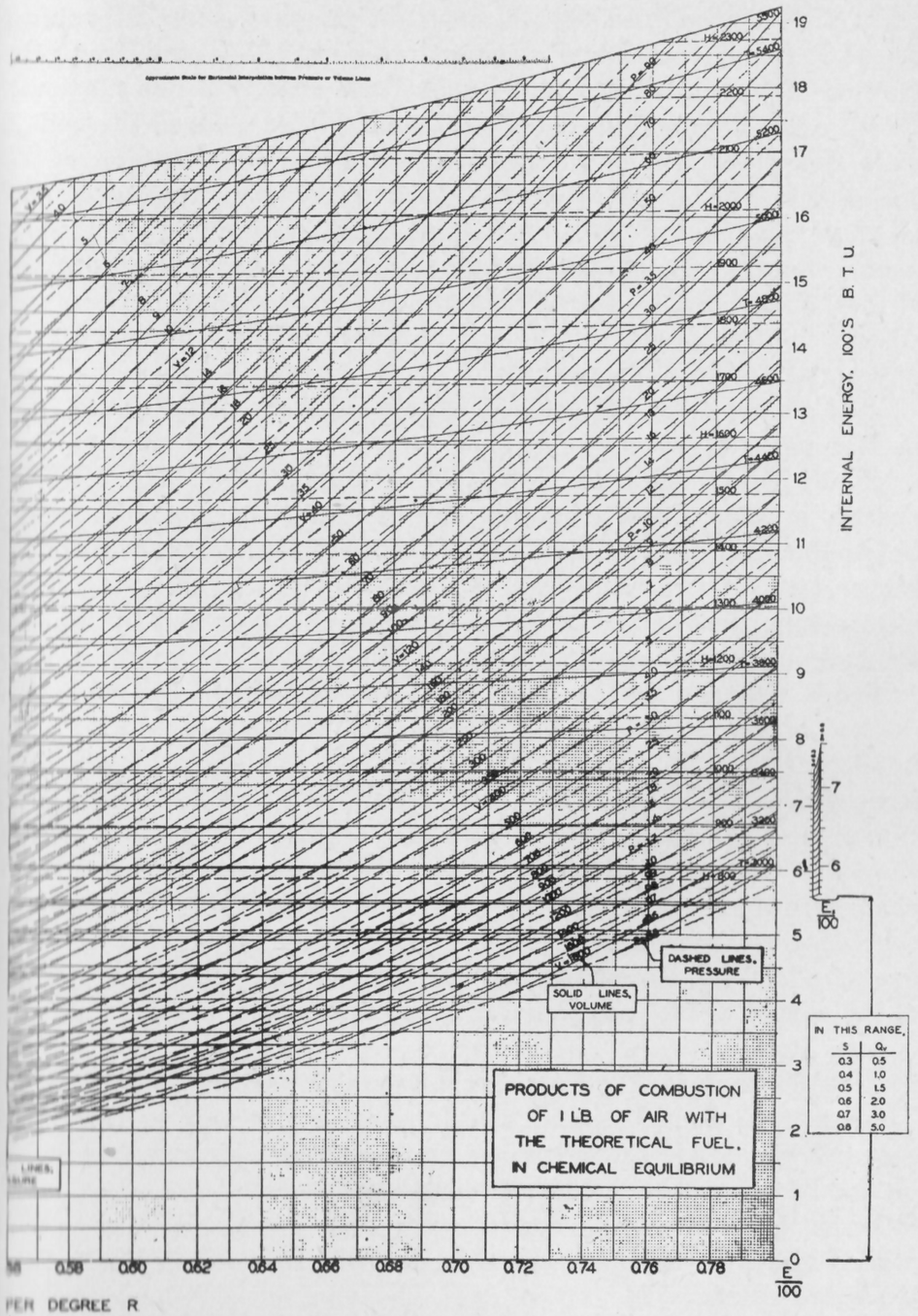


FIG. 41. A Hottel chart giving the thermodynamic properties of the products of combustion of the chemically correct mixture of octene and air. (*McCann, NACA* RB 3G28, 1943.)



On it are superimposed lines of constant pressure, constant volume, constant temperature, and constant enthalpy. The validity of this method of presenting the data can be seen readily if one particular pressure and temperature condition is considered. Since the fuel-air ratio and the weight of charge present are fixed, the amount of dissociation and hence the volume are fixed at any given pressure. With all these quantities defined for that point, it follows that the internal energy, the entropy, and the enthalpy are also defined. Thus any two of these six functions—entropy, internal energy, enthalpy, temperature, specific volume, and pressure—will suffice to define the other four at any particular point on the chart.

The values given for internal energy, enthalpy, and entropy are in Btu per unit weight of charge, *i.e.*, a quantity of mixture that originally contained 1 lb of air before combustion. The internal-energy, enthalpy, and entropy values include the chemical energy obtainable from the combination of the dissociated components in the higher temperature range, where dissociation is appreciable. They also include the chemical energy that could be obtained by burning the carbon monoxide and hydrogen in the products of combustion of mixtures richer than the chemically correct value. By using this method of representation, the change in internal energy during combustion as given by the chart is constant for any given fuel-air ratio irrespective of the temperature and pressure at the end of the combustion process as well as the amount of dissociation. This basis greatly simplifies the use of the charts, for no allowance need be made for changes in the amount of dissociation during an isentropic expansion—the chart already includes these effects. Thus the value for the internal energy added by combustion, E_c, for the chart is simply the product of the weight of fuel per pound of air and the lower heating value for octene, 19,190 Btu/lb.

Use of Charts for Idealized-cycle Calculations for a Diesel Engine. The isentropic compression in a piston-type engine may be calculated readily from data in the air tables, while the isentropic expansion must be handled by using the Hottel charts. The detail procedure to be followed for an idealized constant-volume cycle corresponding to that carried out in a high-speed diesel engine will illustrate one use of the table and charts.

Consider a constant-volume cycle in which the pressure and temperature at the beginning of the compression stroke are 12.4 psia and 650°R, respectively, the compression ratio is 13.9:1, the fuel-air ratio is 0.0543, and the amount of residual exhaust gas is negligible. As in

the example given previously for the air tables, the calculations may be most conveniently carried out on the basis of 1 lb of air. The solution of this problem is given in Table 7. The first column of such a table may be filled in with the given data and with values from the air tables. The specific volume may be computed from the $PV = WRT$ relation. The volumetric compression ratio of 13.9:1 defines both

Table 6. Data for Use with the Hottel Charts at the Back of the Book

Data on Charts for Combustion Products

Per cent theoretical fuel	Fuel-air ratio	Quantity of charge to which chart values for volume, enthalpy, etc., apply			Energy released, by combustion, Btu*	
		Wt air in mixture before combustion, lb*	Wt fuel in mixture before combustion, lb*	Total wt combustion products, lb	E	H
80	0.0543	1.0000	0.0543	1.0543	1040	1037
90	0.0611	1.0000	0.0611	1.0611	1170	1167
100	0.0677	1.0000	0.0677	1.0677	1300	1297
110	0.0746	1.0000	0.0746	1.0746	1428	1425
120	0.0814	1.0000	0.0814	1.0814	1557	1554

Factors for Conversion of Thermodynamic Data for Air Alone from Air Tables to Data for Unburned Fuel-air Mixtures

Per cent theoretical fuel	80	90	100	110	120
Enthalpy of fuel vapor in per cent of the enthalpy of the air alone	11.2	12.4	13.6	14.8	16.0
Internal energy of fuel vapor in per cent of the internal energy of the air alone	18	20	22	24	26

* Values are for fresh mixtures undiluted by exhaust gas.

the specific volume and the relative volume at the end of the compression stroke. The new relative volume may be located in the air tables, and the corresponding temperature, relative pressure, and internal energy determined. At this point, 0.0543 lb of fuel is injected and burned. The heat released is then the product of the fuel-air ratio and the lower heating value of octene, viz., $0.0543 \times 19{,}190 = 1042$ Btu. This may be added to the internal energy at the end of the compression stroke to give the internal energy at the beginning of the expansion stroke. Since the specific volume of the charge in cubic

feet per pound of air remains unchanged, both the internal energy and the specific volume are known. These determine the pressure, temperature, and entropy on the Hottel chart for a fuel-air ratio of 0.0543, viz., 80 per cent of the theoretical fuel. The conditions at the end of the expansion stroke can be determined from the entropy and specific volume. The latter is the same as at the beginning of the compression stroke, while the former is the same as at the beginning of the expansion stroke.

TABLE 7. TABULATED CALCULATIONS FOR A CONSTANT-VOLUME CYCLE FOR A COMPRESSION-IGNITION ENGINE*

Given: Initial pressure = 12.4 psia; initial temperature = 650°R; compression ratio = 13.9:1; fuel-air ratio = 0.0543; fuel = octene.

Condition of charge in cylinder	(1) Start compression stroke	(2) End compression stroke	(3) Start expansion stroke	(4) End expansion stroke	(5) End exhaust stroke
T, °R	650	1732	5070	2760	1980
P, psia	12.4	459	1420	56	12.4
v, ft³/lb air	19.4	1.398	1.398	19.4	65
P_r,	5.478	202.67			
V_r,	2,967	213.5			
U or E†, Btu/lb air	15.49	217.3	1259	531	315
S, Btu/°R	.....	.....	0.4263	0.4263	0.4263

E_c = 1042 Btu
Work out = 1,259 − 531 = 728 Btu/lb
Work in = 217.3 − 15.49 = 201.8 Btu/lb
Net E = 728 − 202 = 526 Btu/lb
Thermal eff = $^{526}/_{1042}$ = 50.5 per cent

$$\text{Imep} = \frac{778 \times 526}{144 \times (19.4 - 1.4)} = 158 \text{ psi}$$

$$\text{Isfc} = \frac{0.0543 \times 2{,}545}{526} = 0.262 \text{ lb/ihp-hr}$$

* No allowance for residual exhaust. Solved with air tables and thermodynamic charts.

† The symbol E in the Hottel charts represents internal energy. It differs from the symbol U in the air tables in that it includes the internal energy of the fuel, not just the air alone.

The temperature of the residual exhaust gas left in the cylinder at the end of the exhaust stroke is often important. Since the gas that remains in the cylinder expands isentropically to force out that which leaves during the exhaust process, the temperature of the residual gas may be found readily from the chart. Its entropy must be the same as that at the end of the expansion stroke, and the pressure in the cylinder must be equal to that in the exhaust manifold, *i.e.*, essentially atmospheric in most cases.

The thermal efficiency, imep, and isfc may be calculated in the manner developed earlier for use with the air tables. Since all the data in the Hottel charts are given for a unit weight of mixture containing 1 lb of air, no correction for the change in weight of charge during the injection of the fuel need be made.

Allowance for Residual Exhaust Gas. The discussion up to this point has been confined to purely ideal cycles. In an actual engine, a certain amount of exhaust gas remains in the clearance volume between the exhaust and intake strokes and mixes with the fresh charge taken in during the subsequent intake stroke. Since the effects of this dilution are very important, it is desirable to take them into consideration in making calculations for an idealized cycle.

The effect of exhaust-gas dilution on the methods of carrying out calculations is not great. Dilution has little effect on the specific heats or the density of the unburned charge at any given temperature and no effect on the composition of the charge after combustion, for the fuel-air ratios for successive cycles are the same. Thus only two factors need be considered, viz., the smaller amount of heat released per unit weight of charge during combustion, and the reduction in the weight of fresh charge taken into the cylinder each cycle because of the displacing effect of the residual exhaust gas.

The heat released by the combustion of the chemically correct or leaner mixtures is equal to that without dilution times the amount of fresh charge per unit weight of the total charge. That is, the value of E_c may be taken as $E_c = 19{,}190(F/A)(1 - f)$, where f is the percentage of residual exhaust gas in the total charge. For mixtures richer than the chemically correct value, the same correction may be applied to allow for the heat released by combustion; but, because of the basis for the charts, another term must be added to allow for the chemical energy available from the combustion of the carbon monoxide and hydrogen in the residual gases. Values for the energy of combustion as a function of the percentage of residual exhaust gas are given in the chart in Fig. 42 for the fuel-air ratios of the charts included at the back of the book.

The effect of residual exhaust gas on the conditions in the cylinder at the beginning of the compression stroke may be taken into consideration by adding some calculations at the beginning of a set such as those in Table 7. To simplify the calculations, assume that the residual exhaust gas remains as a separate layer on top of the piston until

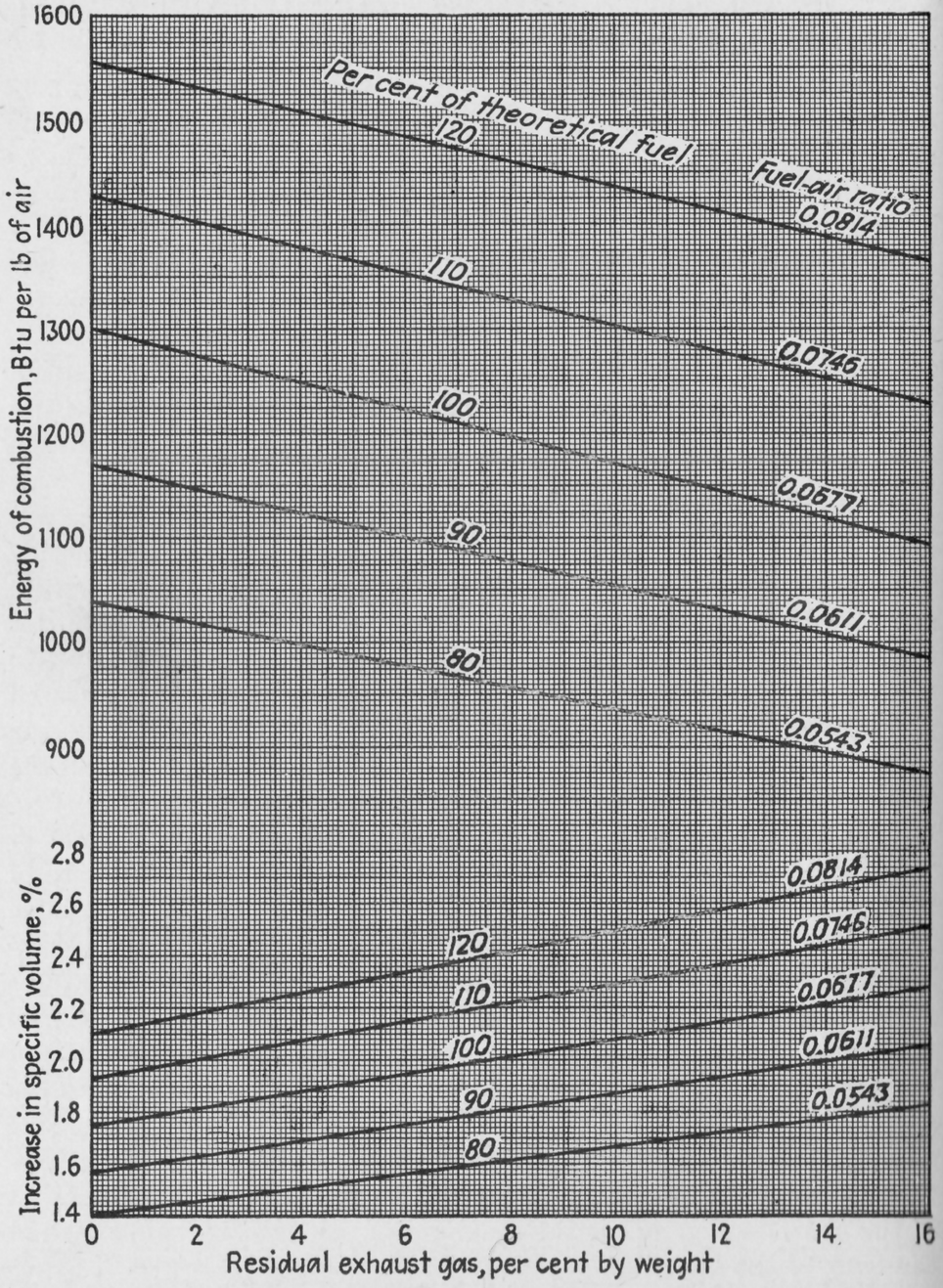


FIG. 42. Chart for converting data from Table 4 to the same basis as the Hottel charts, *i.e.*, from 1 lb of air to the given amount of fuel vapor plus 1 lb of air.

TABLE 8. TABULATED CALCULATIONS FOR A CONSTANT-VOLUME CYCLE FOR A COMPRESSION-IGNITION ENGINE*

Given: Initial pressure = 12.4 psia; initial temperature = 650°R; compression ratio = 13.9:1; fuel-air ratio = 0.0543; fuel = octene.

Condition of charge in cylinder	(*a*) Residual exhaust gas	(*b*) Fresh charge in intake manifold	(1) Start compression stroke	(2) End compression stroke	(3) Start expansion stroke	(4) End expansion stroke	(5) End exhaust stroke
T, °R	2070	650	688	1819	5060	2810	2010
P, psia	12.4	12.4	12.4	456	1320	56	12.4
v, ft³/lb air	65	19.4	20.53	1.476	1.476	20.53	65
P_r,		5.478	6.691	246.2			
V_r,		2,967	2,571	184.8			
E, Btu/lb air	325	15.49	22.06	235.0	1255	528	
S					0.4300	0.4300	0.4300
H, Btu/lb air	463	60.04	69.2				463

$$\text{Weight ratio of residual exhaust gas to total charge} = \frac{1(1/v_a)}{12.9\dfrac{1}{v_b} + 1\dfrac{1}{v_a}}$$

$$= \frac{1\frac{1}{65}}{12.9\dfrac{1}{19.4} + 1\frac{1}{65}} = 0.0225$$

H of fresh charge = $0.9775 \times 60.04 = 58.8$ Btu
H of residual = $0.0225 \times 463 = 10.4$
H of mixture = 69.2 Btu
$E_c = 1020$ Btu
Work out = $1{,}255 - 528 = 727$ Btu
Work in = $235 - 22 = 213$
Net E = 514 Btu

$$\text{Thermal eff} = \frac{514}{1{,}020} = 50.4 \text{ per cent}$$

$$\text{Imep} = \frac{514 \times 778}{144(20.53 - 1.476)} = 145.7 \text{ psi}$$

$$\text{Isfc} = \frac{0.0543 \times 2{,}545 \times 0.9775}{514} = 0.263 \text{ lb/ihp-hr}$$

* With allowance for residual exhaust. Solved with air tables and thermodynamic charts.

the end of the intake stroke. The temperature at the end of the mixing process after the intake stroke may then be determined by adding the product of the weight of fresh charge and its enthalpy to the product of the weight of residual and its enthalpy. Dividing that sum by the total weight of charge gives the enthalpy of the mixture. Since the mixing occurs at constant pressure, the pressure and the enthalpy of

the mixture define the conditions at the beginning of the compression stroke. A set of calculations including these effects is given in tabular form in Table 8 for a cycle similar to that of Table 7. Note that the enthalpy and not the internal energy must be used for these mixing calculations because they are for a constant-pressure mixing process rather than an isentropic nonflow compression or a constant-volume heating process. The residual exhaust data must be estimated.

In comparing the results of the calculations in Tables 7 and 8 it is evident that exhaust-gas dilution of the fresh charge causes a marked drop in imep but has little effect on either the thermal efficiency or the

TABLE 9. TABULATED CALCULATIONS FOR A CONSTANT-VOLUME CYCLE FOR A SPARK-IGNITION ENGINE*

Given: Initial pressure = 12.4 psia; initial temperature = 650°R; compression ratio = 6.0:1; fuel-air ratio = 0.0543; fuel = octene.

	(*a*)	(*b*)	(1)		(2)		(3)	(4)	(5)
			Start compression stroke		End compression stroke				
	Residual exhaust gas	Fresh charge intake manifold	Air only	Corrected for fuel	Air only	Corrected for fuel	Start expansion stroke	End expansion stroke	End exhaust stroke
T, °R	2200	650	764	764	1493	1493	4820	3270	2400
P, psia	12.4	12.4	12.4	12.4	.....	145.3	490	56	12.4
v, ft³/lb air	65	19.4	22.8	23.15	.....	3.86	3.86	23.15	73
P_r	.....	5.478	9.69	.....	113.5				
V_r	.....	2,967	1,971	.....	328.5				
E, Btu/lb air	.....	15.49	35.3	41.6	169.7	200.2	1178	657	423
S	.....	.....	.....	.....	.....	.....	0.4855	0.4855	0.4855
H, Btu/lb air	520	60.04	87.5						592

Weight ratio of residual exhaust gas to total charge $= \dfrac{1(1/v_a)}{5\frac{1}{v_b} + 1\frac{1}{v_a}} = \dfrac{1/6.5}{5\frac{1}{19.4} + 1/6.5} = .06$

H of fresh charge = 0.94 × 60.04 = 56.4 Btu
H of residual = 0.06 × 520 = 31.1
H of mixture = 87.5 Btu
E_c = 978 Btu
Work in = 200.2 − 41.6 = 158.6 Btu
Work out = 1,178 − 657 = 521
Net E = 362 Btu

Thermal eff $= \dfrac{362}{978} = 37.0$ per cent

Imep $= \dfrac{362 \times 778}{144(23.15 - 3.86)} = 101.2$ psi

Isfc $= \dfrac{0.0543 \times 2{,}545 \times 0.94}{362} = 0.359$ lb/ihp-hr

* Solved with air tables and thermodynamic charts.

isfc. This is as one would expect from the basic expression for the efficiency of the cycle given in Eq. (1); for the compression ratio remains constant, and the specific heat of the working fluid is not greatly affected by dilution.

Use of Charts for Idealized-cycle Calculations for a Spark-ignition Engine. The same procedure as that outlined above for a diesel-engine cycle may be followed for a spark-ignition-engine cycle except that the values obtained from the air tables for the compression stroke should be corrected for the presence of fuel vapor. Since the charts for the products of combustion are for 1 lb of air plus the weight of fuel given by the fuel-air ratio, the data from the air tables for the compression stroke may be corrected to the same basis as the charts by simply adding the increments introduced by the fuel vapor to the data for the air alone. The corrections can be applied most conveniently by considering the fuel vapor as being added to the air at constant pressure and temperature. Taking the conditions at the beginning of the compression stroke as an example, the pressure and temperature remain unchanged by the addition of the fuel vapor, while the specific volume in cubic feet per pound of air and the internal energy in Btu per pound of air are both increased. The amount of these increments can be easily determined from Table 6 and from the chart in Fig. 42, which also includes the effects of residual exhaust gas.

An example illustrating the application of these corrections to an idealized spark-ignition-engine cycle is shown in Table 9. With the exception of the corrections to the specific volume and the internal energy for the compression stroke, the procedure is the same as in Tables 7 and 8.

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Problems

1. Consider three large cylinders, each of which has a volume of 15.0 ft³ above a piston. The first contains 1.0 lb, the second 3.0 lb, and the third 20.0 lb of air. The temperature of the air in each case is −60°F.

Determine the air pressure and temperature in each cylinder for each of the following cylinder volumes if the air is compressed isentropically (assume $k = 1.4$) to 10.0, 5.0, and 1.0 ft³. Compute the ratio of the final to the initial volume and the final to the initial pressure for each case, *i.e.*, the relative volume and relative pressure. Tabulate and compare the values for the temperature, relative volume, and relative pressure for each condition.

2. Determine the pressure, temperature, and volume of the charge at the end of the compression stroke, the beginning of the expansion stroke, and the end of the expansion stroke for a constant-volume cycle carried out with 1 lb of air. Take the compression ratio as being 7.0:1, the initial temperature and pressure as 60°F and 29.92 in. Hg absolute, the heat added by combustion as 1,190 Btu/lb of working fluid, and use the air tables for the solution. Also, determine the work output of the cycle in Btu per pound of air, the thermal efficiency, the imep, and the isfc for the cycle (assuming a fuel-air ratio of 0.0677). Plot *P-V* diagrams on both log-log and linear coordinates.

3. Determine the pressure, temperature, and volume of the charge at the end of the compression, the beginning of the expansion, and the end of the expansion for a constant-pressure (gas-turbine) cycle carried out with 1 lb of air. Take the pressure ratio as 6.0:1, the initial temperature and pressure as 60°F and 29.92 in. Hg, the heat added as 200 Btu/lb, and use the thermodynamic data in the air tables. Also, determine the work output of the cycle in Btu per pound of air and the thermal efficiency. Plot *P-V* diagrams on both log-log and linear coordinates.

4. Determine the pressure, temperature, and specific volume of the charge at the four principal points in a constant-volume cycle for a compression-ignition engine. Take the compression ratio as 15.0:1, the initial temperature and pressure as 520°R and 14.7 psia, the fuel as octene, and the fuel-air ratio as 0.0677. Use the air tables for the compression and the Hottel charts for the expansion process. Determine the net work in Btu per pound of air, the thermal efficiency, imep, and isfc for the cycle.

5. Repeat Prob. 4, making allowances for the residual exhaust in the clearance volume.

6. Make a set of calculations similar to those in Prob. 5 for a constant-volume cycle carried out in a 7.0:1 compression-ratio spark-ignition engine with fuel vapor in the air charge at the beginning of the compression stroke.

7. Superimpose the data from Prob. 6 on the *P-V* diagrams plotted for Prob. 2, and compare.

8. Repeat the calculations of Prob. 5 for a series of compression ratios, viz., 4.0:1, 5.5:1, 10:1, 14:1, and 18:1. Plot curves for the thermal efficiency, imep, and isfc vs. compression ratio.

9. Repeat the calculations of Prob. 6 at a constant compression ratio of 7.0:1 for a series of fuel-air ratios, viz., 0.0543, 0.0611, 0.0677, 0.0746, and 0.0814. Plot curves for the thermal efficiency, imep, and isfc vs. fuel-air ratio.

CHAPTER IV

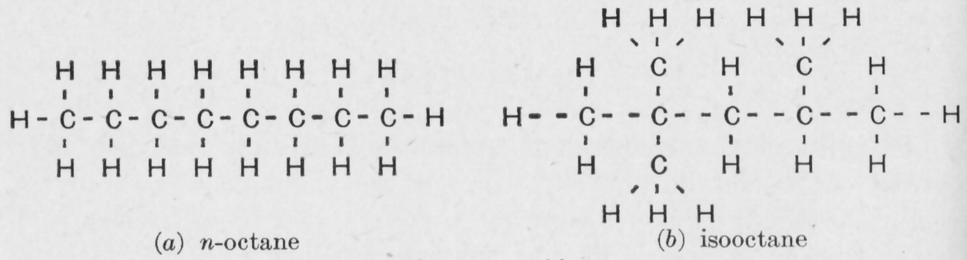
COMBUSTION

In spite of the tremendous amount of research that has been carried out on flames and explosions, these phenomena are still not well understood. Experts working in government organizations such as the Bureau of Standards, the Bureau of Mines, and the National Advisory Committee for Aeronautics, in private research organizations such as General Motors Research and the Ethyl Corp., and in educational institutions such as Massachusetts Institute of Technology and the University of Wisconsin have devoted years to studies of just particular types of combustion. Publications covering certain phases of the work of many of these men are given as references at the end of the chapter. Combustion problems are so complex and so broad in scope, however, that it is doubtful whether many of these experts would agree completely in their explanations or even in their descriptions of the principal types of combustion. In treating so involved a subject, this text aims to present no more than some simplified basic working concepts which can be modified and amplified later through experience and further study of the literature.

COMBUSTION CHEMISTRY

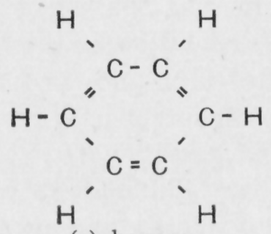
Structure of Fuels. An elementary knowledge of the structure of the hydrocarbon compounds in fuels is essential to an understanding of the peculiarities in the behavior of various types of fuel in combustion engines. Since a section on this subject is included in the chapter on fuels, it will simply be pointed out here that the arrangement of the carbon atoms in a molecule has a very important effect on the susceptibility of that molecule to oxidation. Long straight-chain paraffins such as that indicated in Fig. 43*a* ignite very readily and have low ignition temperatures. A branched-chain paraffin, or isoparaffin, having the same number of carbon and hydrogen atoms as the molecule in Fig. 43*a* but with the atoms arranged to give a more compact structure is indicated in Fig. 43*b*. Such a molecule is much more resistant to oxidation, as indicated by a spontaneous ignition temperature of 1350°F as compared with a similar value of 880°F for the long straight-chain paraffin. Hydrocarbons containing a benzene

ring (see Fig. 43*c*) are called *aromatics* and have ignitability characteristics quite different from those of either straight-chain or branched-chain paraffins.

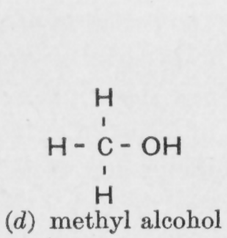


(*a*) *n*-octane

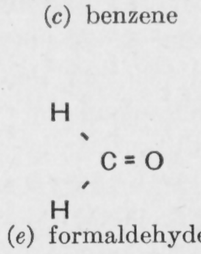
(*b*) isooctane



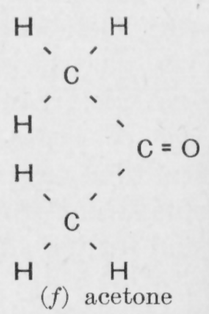
(*c*) benzene



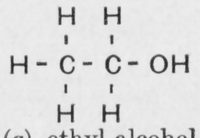
(*d*) methyl alcohol



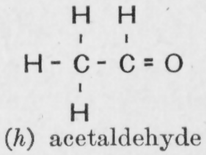
(*e*) formaldehyde



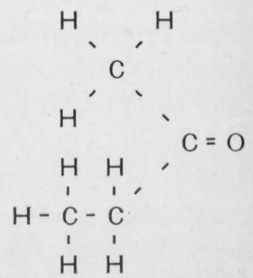
(*f*) acetone



(*g*) ethyl alcohol



(*h*) acetaldehyde



(*i*) methyl ethyl ketone

FIG. 43. Hydrocarbon structures representative of types important in combustion reactions.

Some types of hydrocarbon containing oxygen play important roles in combustion because they may be formed as intermediate compounds when a fuel is oxidized. The most important of these are alcohols, aldehydes, and ketones, the structures of which are indicated in Fig. 43. The aldehydes play particularly important roles and are

often found in the combustion products. The objectionable odors in engine exhaust gas, for example, are caused by aldehydes.

Mechanism of Combustion. Combustion, man's most important source of energy, is still something of a mystery in many ways. The medieval notion that there is something so unique in a flame that a special quantity must be present was in some ways closer to the truth than most people realize. Many simple experiments have been devised to show that simply raising the temperature of a combustible gas mixture will not necessarily make it burn—a certain type of chemical activity peculiar to a flame must be initiated.[1] While the high speed and complexity of the reactions make analysis difficult, it has been demonstrated that even so apparently simple an oxidation process as that in an oxygen-hydrogen flame does not and cannot occur to any appreciable extent by the direct combination of oxygen and hydrogen molecules but rather must take place as the result of a series of intermediate reactions.[1] Thus, while the conventional chemical equation for the reaction is

$$2H_2 + O_2 = 2H_2O$$

this represents an oversimplification as it serves only to relate the initial composition to the products of combustion. While the intermediate reactions might be dismissed by some as being of only academic interest, this is really not the case, for some conception at least of their occurrence is essential to an understanding of many practical combustion problems. Although the precise form of these intermediate reactions varies with temperature, fuel-air ratio, etc., one likely series is as follows:

$$H_2 + OH = H_2O + H$$
$$H + O_2 = OH + O$$
$$O + H_2 = OH + H$$

In examining these equations, two important features are evident. First, a free active ion of OH, H, or O is necessary to initiate the series. In the second place, once such an ion enters into a reaction, it produces *both the oxidation product and a number of new active ions.* A reaction of this sort is called a *branched-chain* reaction because the original active ion produced a number of new active ions. (In this case the original OH ion produced three new active ions.) Each of these will in turn tend to start a new chain of reactions, and thus the reaction rate in the gas as a whole will tend to increase rapidly.

A variety of other chain reactions are undobutedly important in

the oxidation of hydrogen. In these, such active unstable peroxides as H_2O_2, HO_2, and NO_2 play the key roles. The predominant type of reaction occurring in a flame depends a great deal on the concentration of the active ions required to produce it. Dissociation of molecules into simpler forms, a phenomenon mentioned in the last chapter, is one of the important factors in this regard. Dissociation tends to occur at an appreciable rate as the temperature is raised, and thus a higher temperature usually means an increase in the number of active ions—although sometimes it may also mean a decrease if other types of reaction begin to predominate. Certain materials on the combustion-chamber wall or present in the gas itself tend to render inactive or destroy active ions quite selectively. The matter is best visualized in terms of the kinetic theory of gases. In a combustible gas mixture, where there are many different types of intermolecular collision occurring, only certain types of collision are likely to result in a reaction, the probability of any given reaction depending on such factors as the temperature and the composition of the burning gases. Under some conditions one type of reaction will be favored, under others another. An extremely complex set of equilibrium relations governs these reactions and determines the rate of each and hence which predominate.[2]

The oxidation of hydrocarbons involves an even more complex group of reactions than the oxidation of hydrogen discussed briefly above. Cracking of the larger molecules to yield hydrogen and simpler forms of hydrocarbon and the partial oxidation of some hydrocarbons to form active intermediate compounds such as aldehydes, alcohols, peroxides, ketones, and organic acids are some of the reactions that contribute to the complexity of the problem. The aldehydes, in particular, appear to play a very important part in hydrocarbon flames. Spectroscopic analysis, for example, always indicates their presence. The oxidation mechanism for paraffinic molecules appears to start with the partial oxidation of the methyl group at the end of the longest straight chain, thus converting the molecule into an aldehyde.[1] The oxidation process then progresses along the chain.

Ignition Temperature. The ignition temperature might be defined on the basis of the preceding discussion as that temperature above which the rate of formation of active ions is greater than the rate of their destruction by impingement on the walls, collision with heavy molecules, or other means. Thus to bring about ignition the temperature may be raised locally, the gas may be ionized with an electric spark, or—in short—the formation of a sufficient number of active ions to initiate a flame must be induced. Once started, the flame will

usually sustain itself by the continuous production of a plentiful supply of active ions.

The Chemically Correct Mixture. The amount of fuel ideally required to react with all the oxygen in a given quantity of air to yield H_2O and CO_2 is always an important reference condition. The fuel-air ratio of such a mixture is called the *chemically correct* or *stoichiometric* fuel-air ratio. While this quantity varies somewhat from one hydrocarbon to another, octane may be taken as fairly representative of the hydrocarbons in gasoline. Disregarding the intermediate compounds present in the flame during combustion and considering only the material initially present or the final products, elementary chemical considerations show that

$$2C_8H_{18} + 25O_2 = 16CO_2 + 18H_2O$$

Since air is 21.0 per cent O_2 and 79.0 per cent N_2 (including the inert gases with the nitrogen), this may be rewritten to give

$$2C_8H_{18} + 25O_2 + 94.05N_2 = 16CO_2 + 18H_2O + 94.05N_2$$

This may again be rewritten in terms of the products of the molecular weights and the number of moles indicated above to give the relative combining weights.

$$228C_8H_{18} + 800O_2 + 2633N_2 = 704CO_2 + 324H_2O + 2633N_2$$

This may be put in a more simple form in terms of relative weights to give

$$0.0665 \text{ fuel} + 1.00 \text{ air} = .0944H_2O + 0.205CO_2 + 0.767N_2$$

The chemically correct fuel-air ratio for an octane-air mixture thus becomes 0.0665.

The chemically correct mixture varies somewhat from one hydrocarbon to another because it depends on the ratio of the weight of hydrogen to the weight of carbon in the compound, *i.e.*, the *hydrogen-carbon ratio*. Hydrocarbons having the general formula C_nH_{2n} yield chemically correct fuel-air ratios of 0.0677. The highest and lowest chemically correct fuel-air ratios for any hydrocarbons commonly found in petroleum fuels are those for benzene and methane, respectively; namely, 0.0754 and 0.0580. Inasmuch as commercial fuels are normally mixtures of hydrocarbons, the chemically correct mixture must be determined from the hydrogen-carbon ratio of the fuel. This may be accomplished by setting up a number of equations relating the hydrogen-carbon ratio of the fuel and the oxygen-nitrogen ratio

of the air to the composition of the products of combustion. These may be solved simultaneously to give a general expression for the chemically correct fuel-air ratio.

$$\text{Chemically correct } \frac{F}{A} = \frac{0.0874\ (1 + \text{hydrogen-carbon ratio})}{1 + 3\ (\text{hydrogen-carbon ratio})} \qquad (9)$$

Since most commercial gasolines and diesel fuel oils in the United States have hydrogen-carbon ratios of very nearly 0.167, the chemically

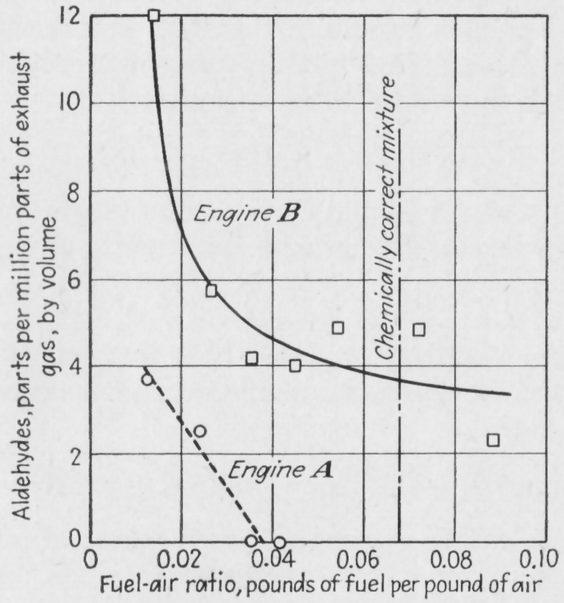


FIG. 44. Effect of fuel-air ratio on the aldehyde concentration in the exhaust gas of two diesel engines. (*Holtz and Elliott, Trans. ASME, Vol.* 63, 1941.)

correct fuel-air ratio may be taken as 0.0677 with negligible error for practical purposes.

Combustion Products. The products of combustion as found in engine exhaust gases consist predominantly of N_2, O_2, H_2O, CO_2, and CO. If relatively lean mixtures (*i.e.*, relatively small amounts of fuel for the amount of air present) are used, appreciable amounts of aldehydes will be present. Since the aldehydes are primarily responsible for the offensive odors in exhaust gas, the curves in Fig. 44 showing the aldehyde content of the exhaust from two representative diesel engines[3] are of interest. Test data for spark-ignition engines also show that the aldehyde concentration is almost negligible for chemically correct mixtures but that it increases rapidly as the mixture is leaned out, apparently as a result of incomplete combustion.[4] If very rich mixtures (*i.e.*, those having relatively high fuel-air ratios)

are used, carbon in the form of smoke will appear. However, for most practical purposes, only the relative proportions of N_2, O_2, CO_2, CO, and H_2 are of importance. These proportions vary widely with the fuel-air ratio[3,5] and, because of dissociation, they also vary somewhat with temperature above about 3000°R.

Since practically all exhaust-gas-analysis work is conducted on the gases after they have been cooled to room temperature and since the relations governing the composition in the dissociation range are very complex, only the composition at temperatures where dissociation is negligible will be considered here. For these conditions it has been found that, for mixtures leaner than the stoichiometric value, the fuel tends to burn completely to CO_2 and H_2O. If the mixture is richer than the stoichiometric value, neither the carbon nor the hydrogen is burned completely. The hydrocarbon molecules crack at temperatures above about 1000°F. Under rich mixture conditions some of the hydrogen and methane formed by this cracking is never burned and leaves with the exhaust. The carbon (except that in the methane) burns to CO and CO_2 with no elemental carbon formed except under very rich mixture conditions in which the burning gases are chilled before the complex combustion reactions have progressed to completion.

While the composition of the exhaust gas from an engine at any given fuel-air ratio varies somewhat with the hydrogen-carbon ratio of the fuel, the curves in Fig. 45 are representative. They give the average composition of the exhaust gases from three spark-ignition engines as a function of fuel-air ratio.[5] A very similar set of data has been obtained for diesel engines.[3]

Three vertical lines are drawn on Fig. 45 to indicate critical or important fuel-air ratios. One is for the chemically correct fuel-air ratio. The other two are for the rich and lean combustibility limits. While it is possible to ignite homogeneous fuel-air mixtures in a bomb at fuel-air ratios outside of the combustibility range indicated in Fig. 45, the time required for combustion is so great that combustibility limits determined in this fashion have little significance from the standpoint of engine operation. The combustibility range indicated in Fig. 45 is significant because a short-duration spark occurring in a mixture leaner than the lean limit or richer than the rich limit will not ignite the fuel vapor. Combustion of mixtures leaner than the lean limit is possible only if the fuel vapor is not uniformly distributed throughout the air. Such a condition prevails in a diesel engine where only in the immediate vicinity of the fuel spray is the mixture in the combustion chamber rich enough to burn. Thus the

diesel can and does normally operate with an average over-all fuel-air ratio that is leaner than the lean combustibility limit. The mixture in a spark-ignition engine on the other hand, must be in the combustibility range if the engine is to run under its own power.

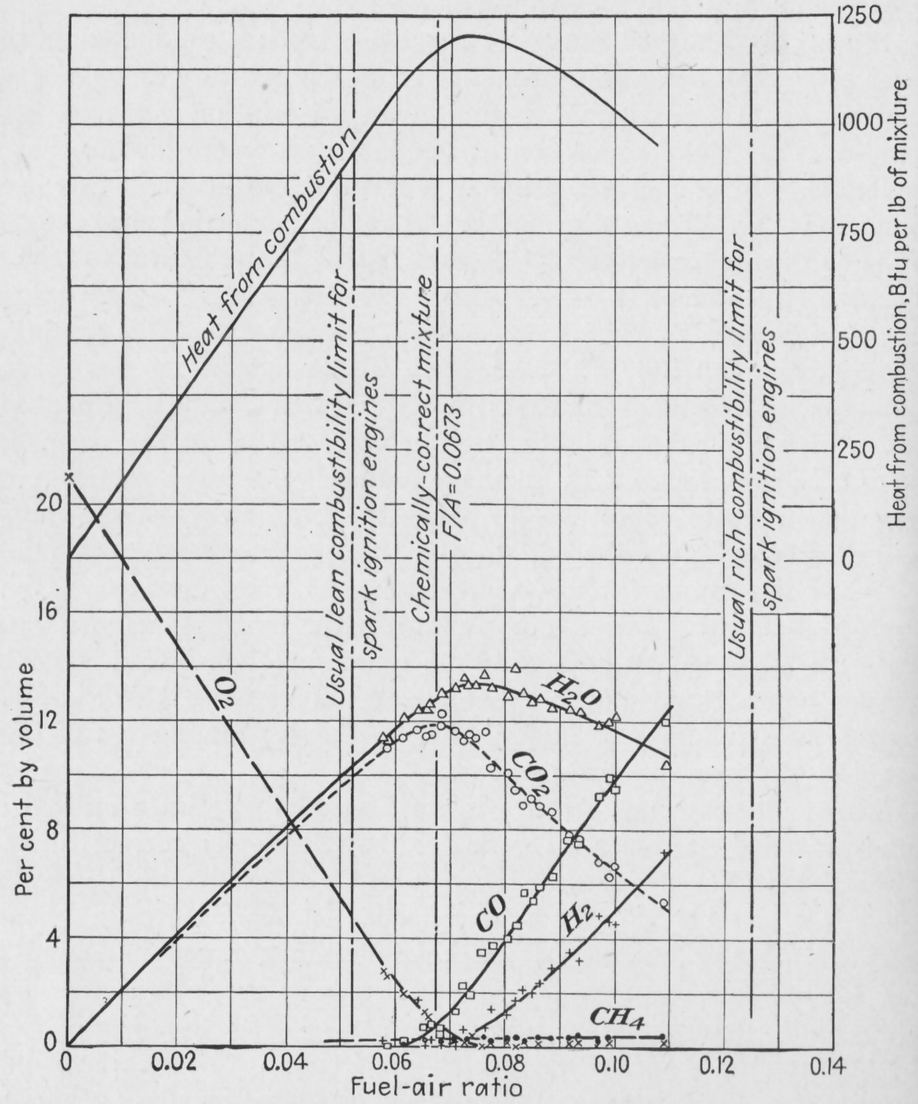


FIG. 45. Effect of fuel-air ratio on the composition of the products of combustion of gasoline and air, and on the heat released in Btu per pound of mixture. Hydrogen-carbon ratio = 0.176. (*Calculated from data of D'Alleva and Lovell.*[5])

Heat Released by Combustion. A number of qualifying factors affect the amount of heat released by combustion.[6] For one thing, the heat of vaporization of the water formed by combustion has never been made available in an internal-combustion engine. A second

factor mentioned previously is dissociation. A third factor is the hydrogen-carbon ratio of the fuel. Since the heating value of hydrogen is 61,045 Btu/lb while that of carbon is 14,520 Btu/lb, the greater the proportion of hydrogen in a fuel, the higher its heating value. However, from the practical standpoint, the fuel-air ratio is in many ways more important than all three of the above items put together. Under favorable lean mixture conditions, practically all the heat ideally available from combustion can be obtained. But if the fuel-air ratio is richer than the chemically correct value, the fuel cannot be burned completely. The magnitude of the loss entailed may be shown by totaling the chemical energy released by combustion for a rich mixture condition, taken from Fig. 45, and comparing it with the chemical energy released by the combustion of the chemically correct mixture. These calculations are shown in Table 10. Since the heat available to raise the temperature of the charge after compression does not include the heat of vaporization of the water, two sets of values are given for the heat released by combustion. One is that for the total amount of chemical energy released, while the other is for the heat available to raise the temperature of the charge. The former value for the stoichiometric mixture is called the *higher heating value* of the fuel, while the latter is the *lower heating value.*

TABLE 10. HEAT RELEASED BY COMBUSTION*
(Calculated for Two Fuel-air Ratios)

Fuel-air ratio	0.0680		0.0100	
Heat from hydrogen, Btu/lb H_2	61,045		61,045	
Hydrogen burned, lb/lb fuel	0.151		0.105	
Heat from hydrogen, Btu/lb fuel		9,220		6,410
Heat from carbon burned to CO, Btu/lb C	4,340		4,340	
Carbon burned to CO, lb/lb fuel	0		0.535	
Heat from carbon burned to CO, Btu/lb fuel		0		2,320
Heat from carbon burned to CO_2, Btu/lb C	14,520		14,520	
Carbon burned to CO_2, lb/lb fuel	0.849		0.314	
Heat from carbon burned to CO_2, Btu/lb fuel		12,310		4,560
Heat of formation of fuel, Btu/lb fuel		−1,360		−1,360
Gross heat released, Btu/lb fuel		20,170		11,930
Water formed, lb/lb fuel	1.359		0.891	
Heat of vaporization of water, Btu/lb water	−970		−970	
Heat required to vaporize water, Btu/lb fuel		−1,317		−865
Net heat available if water is vaporized, Btu/lb fuel		18,853		11,065

* Composition of combustion products taken from Fig. 45.

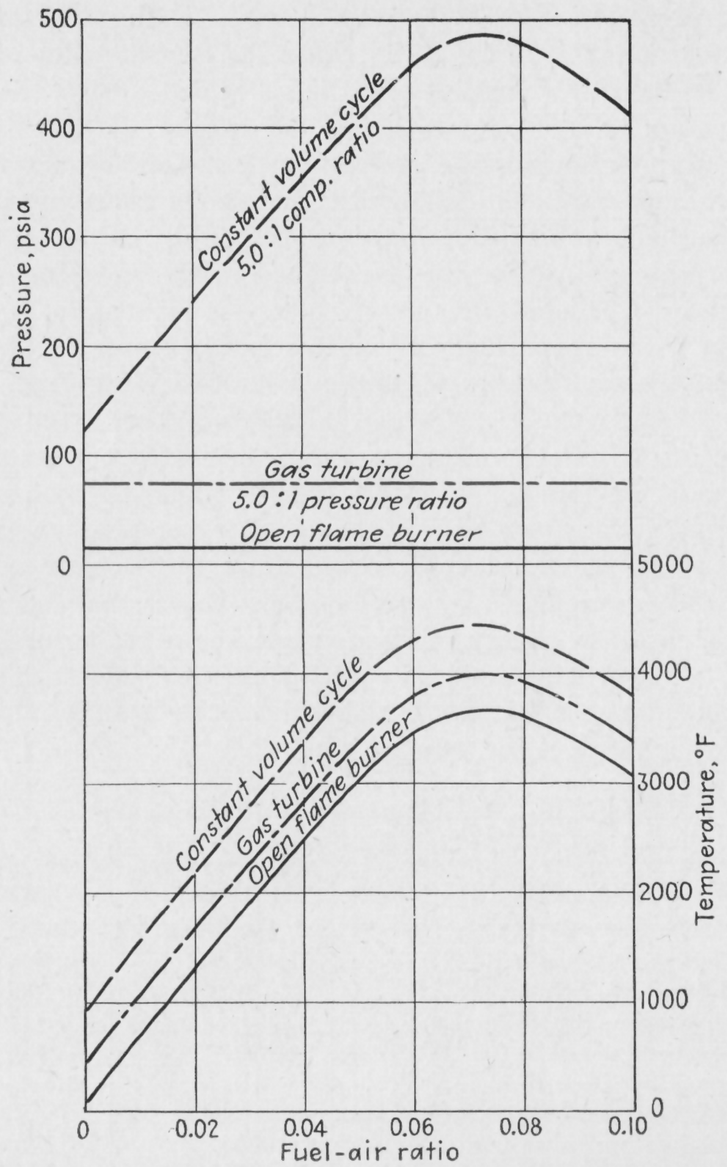


FIG. 46. Effect of fuel-air ratio on the ideal temperature and pressure after combustion for an octene-air mixture in an open flame, a constant-pressure (gas-turbine) cycle, and a constant-volume cycle. Full allowances for variable specific heats, dissociation, etc., were made in the calculations.

Flame Temperature. The heat released by the combustion process usually goes almost entirely into heating the products of combustion, including the nitrogen in the original air charge. The temperature after combustion depends primarily on the amount of heat relased per pound of mixture, which, in turn, is primarily a function of fuel-air ratio (see Fig. 45). In considering the magnitude of the effects

in a particular case, take the condition at the end of the compression stroke in the fuel-air cycle shown diagrammatically in Fig. 35 (page 38). This gave a temperature of 975°R at the beginning of combustion. Using the values given in Table 10 for the heat released for each of the two fuel-air ratios (0.0680 and 0.100), an average specific heat c_v of 0.27, and neglecting the effects of dissociation, it follows that

For $F/A = 0.0680$:

Heat released = 18,853 × 0.0680 Btu/lb air
Heat absorbed by combustion products = $Wc_v(T_2 - T_1)$
$= 1.0680 \times 0.27(T_2 - 975)$

Equating,

Heat released = heat absorbed

$$18{,}853 \times 0.0680 = 1.0680 \times 0.27(T_2 - 975)$$

$$T_2 = \frac{18{,}853 \times 0.0680}{1.0680 \times 0.27} + 975 = 4{,}445 + 975 = 5420°\text{R}$$

For $F/A = 0.100$:

Heat released = 11,065 × 0.100 Btu/lb air
Heat absorbed by combustion products = $1.100 \times 0.27\ (T_2 - 975)$
$T_2 = 3{,}725 + 975 = 4700°\text{R}$

These rough approximations show that the flame temperature falls off only about half as rapidly with an increase in fuel-air ratio as does the heat obtained per pound of fuel because of the greater amount of fuel present per pound of air. Figure 46 was carefully computed with allowances for the variable specific heats and dissociation to show the temperature of the products of combustion as a function of fuel-air ratio for several typical sets of conditions. Experimental data have shown excellent agreement with such calculated curves.[1,7,8]

FORMS OF COMBUSTION IN FLAMES AND EXPLOSIONS

Types of Combustion. Many types of combustion reaction have been investigated. Although only about half of these are found in engines, an examination of their points of similarity and difference is instructive. The types of combustion might be grouped into six main classifications with descriptions as follows:

Open Flames Supplied by a Gas Stream in Which the Flow Is Laminar. The bunsen burner is a familiar example of this type of flame. It is characterized by a thin primary zone in which the oxygen ini-

tially in the mixture combines with the combustible. Additional oxygen from the air surrounding the flame diffuses into the burning gas stream at a uniform rate. Thus further oxidation takes place along any given streamline from the primary combustion zone to the point where combustion is complete or the gases are chilled to a temperature too low for combustion.[1] The primary combustion zone is kept at the burner tip by mixing an inadequate supply of primary air with the combustible so that the velocity of the primary flame is lower than the velocity of the gas stream. The eddies at the tip of the burner where the gas stream emerges into the atmosphere provide a low-velocity zone in which the flame is maintained and from which it can propagate toward the higher velocity core. The flame boundaries are smooth and sharply defined. The ionization from the chemical activity in the flame is responsible for its luminosity.

Open Flames Supplied by a Gas Stream in Which the Flow Is Turbulent. The flame from a plumber's gasoline blowtorch is of this type. These flames are similar to those described above except that admixture of air takes place by turbulence as well as by diffusion so that the combustion rate per unit volume of flame is higher and the eddies in the turbulent stream of burning gases break up the flame boundaries into highly irregular and unstable surfaces.

Flames in Enclosed Quiescent Mixtures or Enclosed Gas Streams in Which the Flow Is Laminar. The most easily visualized flames of this type are those in mine explosions or in laboratory bombs. Since oxygen has already been mixed with the combustible gas, combustion is not delayed by the time required for the admixture of oxygen. Because of this, combustion in any given element of gas mixture is so rapid once it has been touched by the flame that the reactions proceed to completion before the front surface of the flame has progressed more than a microscopic distance beyond the element. Thus the luminous zone of intense chemical activity is only a few thousandths of an inch in thickness. In a chamber containing a quiescent mixture, the flame takes the form of a thin, smooth shell much like a rapidly expanding soap bubble. Once started, the flame progresses radially outward from its origin until it reaches the walls of the chamber and no combustible material remains. The velocity of the flame relative to the gas is moderate—of the order of 10 ft/sec—and depends on the composition of the mixture and its temperature and pressure.

Flames in Enclosed Turbulent Mixtures or Gas Streams. Flames in engine cylinders are of this type and are similar to that described above except that the smooth, thin, shell-like flame is broken up by

turbulence into a highly irregular, unstable, and rather deep zone of activity in which there are a great many pockets of unburned gas, each, in effect, surrounded by an element of thin flame.[8] The velocity of the flame relative to the gas is dependent on the intensity and scale of the prevailing turbulence as well as the pressure, temperature, and composition of the mixture so that it is of the order of two to ten times the velocity of a flame under corresponding conditions in a quiescent mixture.

Autoignition. If the temperature of a combustible-gas mixture is raised sufficiently, combustion reactions will get under way almost simultaneously throughout the mixture. These may progress very slowly, at moderate speeds, or very rapidly, depending on the type of fuel, the composition, temperature, and pressure of the mixture, and the speed with which the temperature is increased. That is, if the temperature of the gas is raised very suddenly to a high value, it is likely that a short period of no apparent activity will be followed by a very rapid reaction. If, on the other hand, the temperature were raised slowly, the reaction would probably begin at a lower temperature and progress at a less rapid rate.[8]

Detonation. The term *detonation* is employed to describe a type of reaction that takes place in a shock wave. The extremely rapid rate of the reactions in fulminate of mercury, TNT, or nitroglycerin is responsible for the violence of the effects of these explosives. The rapidity of the reactions is explained by the fact that all the chemical activity involved takes place in a shock-wave front, where an extreme pressure gradient exists. The same phenomenon has been observed in hydrocarbon gas mixtures both in tubes and in engines.[1,8] The velocity of propagation of the shock wave in gases is very high, being of the order of 2,000 to 8,000 ft/sec.[1,8]

Any one or all three of the last three types of combustion listed above may occur in an engine cylinder. If properly interpreted, information on the other types of combustion is also valuable as a means of gaining an insight into combustion problems in engine cylinders. Similarly, lessons learned from combustion in engine cylinders can be used to advantage in coping with combustion problems in burners such as those for gas turbines.

FLAME MOVEMENT AND CYLINDER PRESSURE DEVELOPMENT IN SPARK-IGNITION ENGINES

Flame Movement. In spark-ignition engines the small flame started at the spark-plug electrodes spreads outward slowly at first

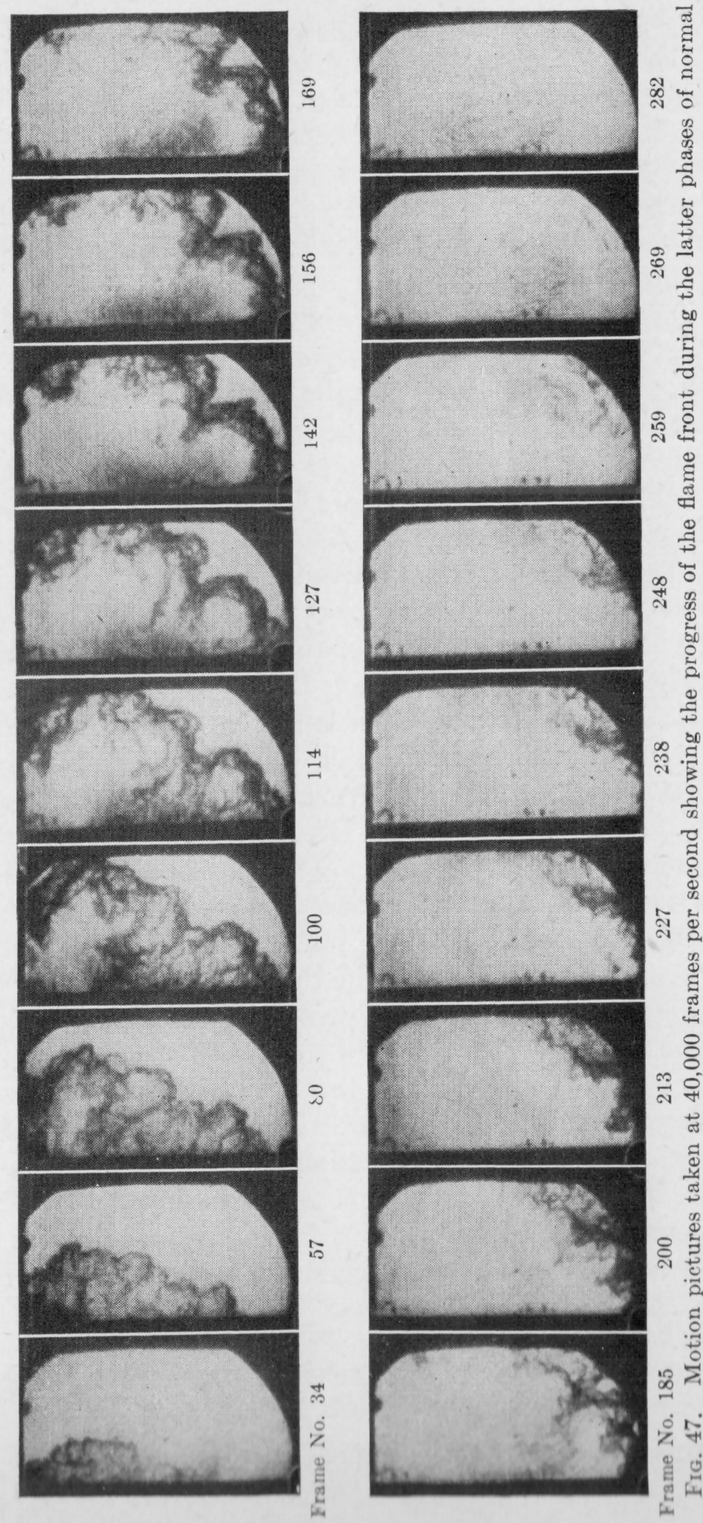
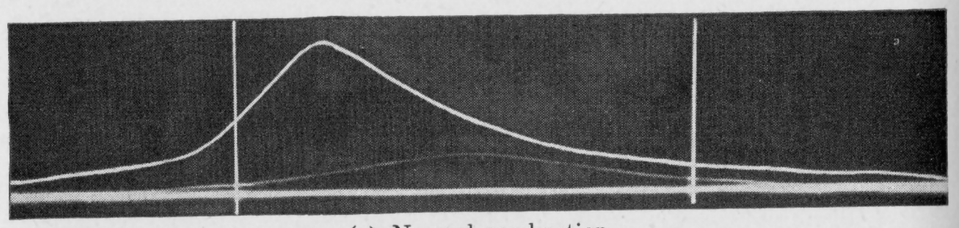


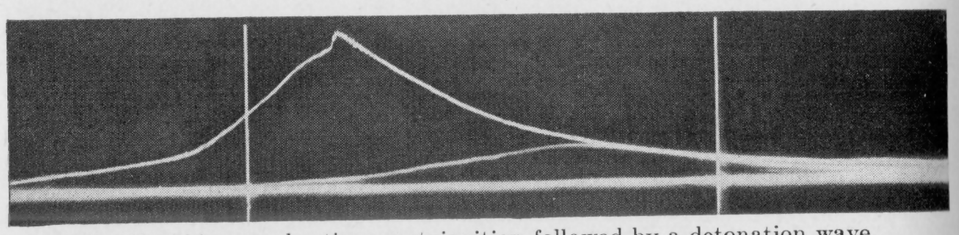
FIG. 47. Motion pictures taken at 40,000 frames per second showing the progress of the flame front during the latter phases of normal combustion in a spark-ignition engine. (*NACA Aircraft Engine Research Laboratory.*)

and then more rapidly, the combustion mechanism being of the fourth type described in the previous section. The progress of the flame front has been photographed with high-speed motion-picture cameras by several different methods.[8,9,10] One excellent set of pictures is shown in Fig. 47.[8] These pictures were obtained by taking Schlieren photographs through a window in a cylinder head. Schlieren photography was employed because the flame in a spark-ignition engine cylinder is only faintly luminous (much like that of a bunsen burner) and hence gives too little light for conventional-type photographs at ultrahigh speeds. Schlieren photographs such as those used in fluid-flow studies are not handicapped in this way. The pictures are taken by directing light toward the camera through the fluid being photographed. Differences in density within the fluid cause differences in refractive index and hence cast a shadow. While it is not evident in Fig. 47 (which was prepared from approximately every fifteenth frame of the film), the motion picture discloses that the entire flame front is extremely turbulent, the effect as a whole being that of a tangled mass of eddies and crosscurrents. Turbulence similar to this but of a smaller scale prevails throughout the combustion chamber. Turbulence is difficult to see in the motion pictures except where the charge is markedly nonhomogeneous, *i.e.*, in the portion that is burning. Fine-grain turbulence both before and during combustion seems to characterize gas conditions in all types of highly developed combustion chamber for both piston engines and gas-turbine burners.

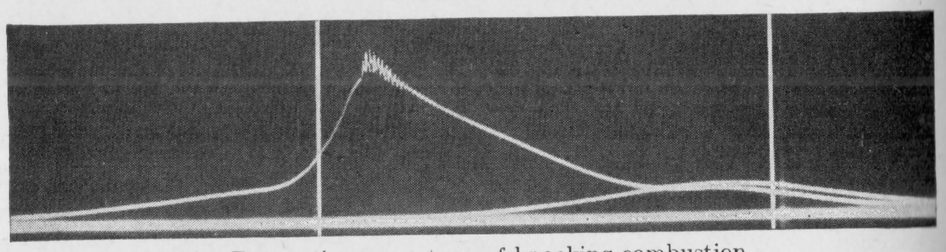
Pressure Development. As the flame moves across the cylinder, the heat released by combustion causes the burned portion of the charge to expand. During the early and middle stages of the process, this expansion causes a large apparent increase in the velocity at which the flame moves with respect to the combustion chamber. Both the unburned and the burned portions of the charge are compressed isentropically as the burning portion of the charge expands, the increase in pressure being directly proportional to the percentage of the charge that has been burned.[8,9,10] Figure 48 gives oscillograms for cylinder pressure as a function of crank angle, *i.e.*, of time. Only the upper trace of each diagram is significant in this instance—the lower traces are for the intake stroke or for nonfiring cycles. Note in Fig. 48*a* (which is for normal combustion) that the spark timing has been selected so that approximately half of the charge is burned before and half after top center. As will be demonstrated later, such a spark setting gives the maximum power and the maximum economy and reduces the peak pressure because the piston has passed top dead



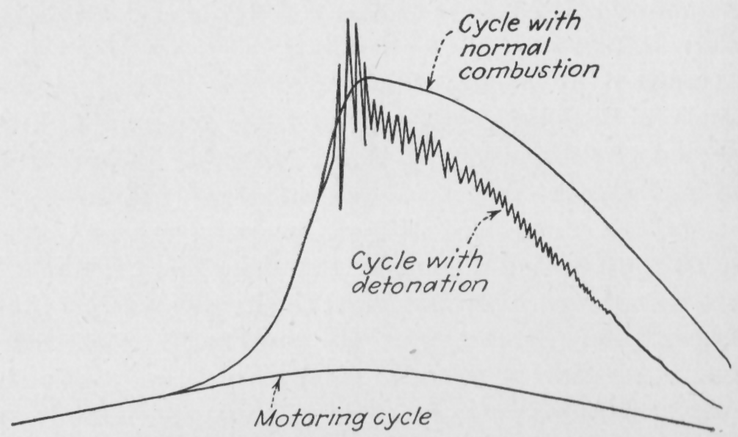
(*a*) Normal combustion.



(*b*) Knocking combustion—autoignition followed by a detonation wave.



(*c*) Detonation-wave type of knocking combustion.



(*d*) Superimposed diagrams for normal and for knocking combustion.

FIG. 48. Pressure-crank angle diagrams for a spark-ignition engine showing three types of combustion. (*NACA Aircraft Engine Research Laboratory.*)

center before combustion has been completed. This and other matters associated with the indicator diagram are covered in Chap. V.

Factors Affecting Flame Velocity. While practically all operating variables have some effect on flame speed, by far the most important is turbulence in the combustion chamber. The effects of this factor are so great that it has been found that the number of degrees of crank rotation required for combustion increases with engine speed only a moderate amount. The reason for this is that turbulence set up in the charge as it is drawn into the cylinder during the intake stroke rapidly increases in intensity with rpm and persists throughout the compression and combustion processes. The type of turbulence that appears to be most desirable is the fine-grained sort indicated in the pictures of Fig. 47 rather than a large-scale swirl about the center of the combustion chamber.

Another important factor affecting flame speed is fuel-air ratio. The maximum flame speed is obtained with a fuel-air ratio somewhat richer than the chemically correct value, very rich or very lean mixtures yielding flame speeds as much as 30 or 40 per cent lower. In fact, as the rich or lean combustibility limits are approached, flame movement becomes very irregular and unpredictably slow. While other factors such as pressure, gas temperature, humidity, and percentage of residual exhaust gas affect flame velocity,[11,12] their effects are not large and their relationships are so complex when all varieties of engine are considered that a detailed discussion of them cannot be included here.

Temperature Variations in the Products of Combustion. One important effect resulting from the nonsimultaneous burning of the charge is that of temperature variation in the products of combustion. The first part of the charge to burn experiences a large temperature rise at constant pressure, followed by an adiabatic compression. The last portion of the charge to burn, however, follows a different process, for it is first compressed adiabatically and then burned at constant pressure. The resulting temperatures for the two elements are different; for although the amount of heat added during the constant-pressure burning processes is essentially the same for both, the temperature rise during the adiabatic compression process is considerably greater for the first element to burn than for the last. This can be seen easily from the elementary thermodynamic relation for the final temperature T_2 for an isentropic compression.

$$T_2 = T_1 \left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}}$$

or

$$\text{Temperature rise} = T_2 - T_1 = T_1 \left[\left(\frac{P_2}{P_1} \right)^{\frac{k-1}{k}} - 1 \right]$$

Since the quantity within the brackets depends only on the pressure ratio and since that quantity is the same for both the first and the last elements to burn, it becomes evident that the temperature rise during that portion of the process involving isentropic compression is directly proportional to the absolute temperature at the beginning of the compression. Thus the temperature of the charge after the whole combustion process has been completed decreases progressively from the first to the last part of the charge to have been burned. Therefore, after combustion has been completed, the temperature of the gas in the vicinity of the spark plug is higher than that of any other portion of the charge. This is one of the reasons why the spark-plug electrodes tend to present one of the hottest surfaces in the combustion chamber.

Preignition. The discussion up to this point has been concerned with what might be called *normal combustion*. Under certain engine operating conditions, however, other phenomena may be encountered. One of these, referred to as *preignition*, may occur when some portion of the combustion-chamber surface such as the spark-plug electrodes, an exhaust valve, or a glowing piece of carbon acts as a hot bulb and ignites the fuel-air mixture in its vicinity. Such a hot spot originates a flame front of the same character as that from an electric spark.[13] The important difference between the two is that the timing of the origin of such a flame front is both erratic and uncontrollable. Under some conditions the point at which preignition occurs may tend to fall a little earlier each cycle until a point is reached at which combustion will be completed so early on the compression stroke that the engine will tend to stall. The reason for this may be deduced from the discussion in the preceding section on the temperature rise accompanying an isentropic compression. If combustion occurs before the compression, the final temperature will be much higher than if combustion follows the compression. Thus, if preignition once starts, the ignition tends to occur progressively earlier on the compression stroke and thus yields still higher gas temperatures because the pressure rise following combustion becomes greater. The higher gas temperatures coupled with the increase in the duration of exposure of the combustion-chamber walls to the hot gases tend to increase further the temperature of the hot spot.

In general, preignition will occur in only one or two cylinders of a multicylinder engine at a time so that the engine will continue to run in spite of preignition in some cylinders. These cylinders will give very little power or even tend to "kick back," thus causing one form of "rough" operation (*i.e.*, operation characterized by irregular vibration). In such cases, the hot spot in the preigniting cylinder may even ignite the fresh charge coming into that cylinder during the intake stroke, thus causing a backfire through the intake manifold. The hot spot causing preignition may become so hot in an overheated engine that the engine will continue to fire when the ignition is turned off. When this occurs, it is termed *afterfiring*.

Aside from the very undesirable operating characteristics attending preignition, engine parts may sometimes be severely damaged by either the high pressures or the high temperatures that are induced. The hot spots may become so severely overheated that the metal will be melted and "burned" away. The nature of this action is discussed in greater detail at the end of the next section.

Knocking Combustion. The conditions of high temperature and pressure that prevail in the last portion of the charge to burn just before it is reached by the flame front may bring about either autoignition or detonation. If this happens, the energy of combustion in the last part of the charge is released at a rate as much as a hundred times greater than that for normal combustion. The resulting abrupt and violent increase in pressure produces an audible knock. While all types of knocking combustion in both spark-ignition and diesel engines are often called "detonation," a true detonation wave as described on page 87 is not always responsible for the audible knock.

The indications are that the very rapid combustion which is characteristic of knock occurs only after the unburned portion of the charge has changed somewhat in character, *i.e.*, after certain reactions have occurred in the unburned charge under conditions of high temperature and pressure. Spectroscopic data taken in the extremely short time available show that aldehydes are present in the unburned charge before knocking occurs.[1] Other evidence supporting the concept that reactions precede knocking combustion arises from the fact that there is a definite time element involved. If operating conditions can be changed so that the cylinder charge is burned more quickly, knock may be avoided.

Figure 49 shows frames taken from ultrahigh-speed motion pictures of the autoignition type of knocking combustion, while Fig. 50 shows similar pictures of a detonation wave. A careful examination and

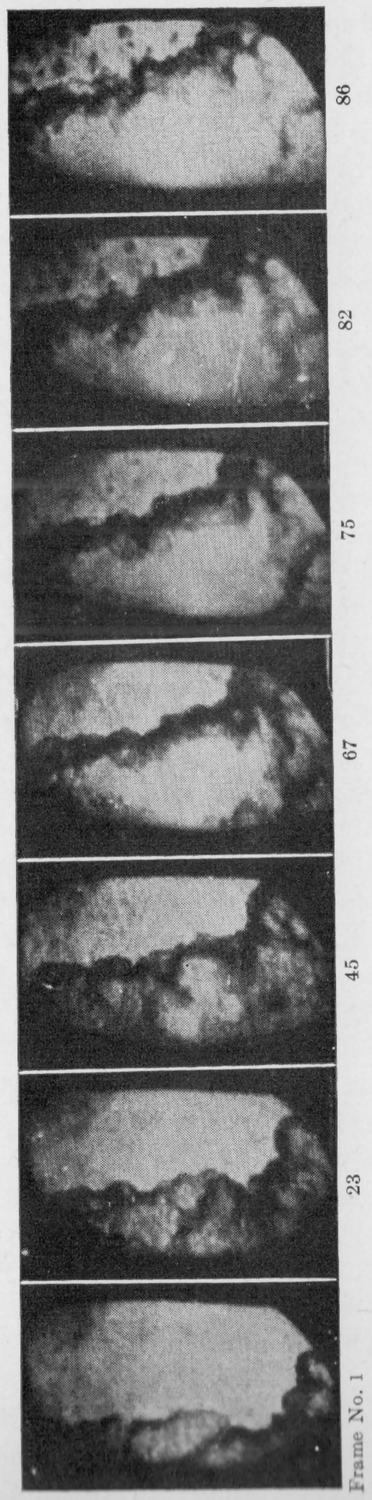


FIG. 49. Motion pictures taken at 40,000 frames per second showing the flame development for the autoignition type of knock followed by a detonation wave in a spark-ignition engine. (*NACA Aircraft Engine Research Laboratory.*)

comparison of these two sets of pictures is very instructive, especially if the progress of the flame front is carefully correlated with time as indicated by the frame number. In the cycle depicted in Fig. 49, for example, combustion progressed normally until autoignition began, as evidenced by the dots appearing in the unburned portion of the field in frame 75 of Fig. 49. Frame 82 and subsequent frames show that these initial reaction centers had spread and new centers had formed rapidly so that by frame 92 the entire field in what had been the unburned portion of the charge was covered with centers of activity. A detonation wave then started in the upper right-hand corner of frame 94. This wave completely traversed the charge by frame 96, burning all that which had not been burned up to the time of its passage. As in this instance, a detonation wave often accompanies autoignition.[8]

The detonation wave in the 200,000-frame-per-second pictures of Fig. 50 is difficult to see in a series of stills although the activity can be distinguished rather well if the pictures are projected on a screen in slow-motion form. The detonation wave started from the origin indicated by the arrows in the flame front in frame 6 and progressed outward in all directions through both the burned and the unburned charge. The dark striations in frame 18 were caused by large-amplitude movements of carbon particles in the detonation wave after it had been reflected back from the combustion-chamber wall.

By counting the number of frames taken during the knocking portion of the combustion process, note that with the autoignition type of knock the time required for combustion of the knocking portion of the charge was only 21/40,000 sec, whereas the time required to burn the corresponding amount of charge under normal combustion conditions was 182/40,000 sec, a difference in reaction rate of about 10 to 1. The difference in combustion rate was even greater when the bulk of the knocking combustion occurred in a detonation wave. It can be seen from Fig. 50 that for such a condition the combustion process occurred in less than 12/200,000 sec., a reaction rate nearly a hundred times that of normal combustion. Cylinder-pressure diagrams for the three types of combustion of Figs. 47, 49, and 50 are shown in Fig. 48. The differences in the relative combustion rates are clearly indicated by the rates of pressure rise in each instance. Note the rapid rate of pressure rise in Fig. 48*b* from the portion of the charge that burned under autoignition conditions and the small high-frequency pressure fluctuations set up by the last tiny portion that burned in a detonation wave. Since a far larger portion of the charge

burned in the detonation wave in the cycle of Fig. 48*c*, the amplitude of the high-frequency pressure waves is much greater. Note, too, that the cylinder pressure curve is virtually vertical during combustion of the knocking portion of the charge in Fig. 48*c*, whereas in the same range in Fig. 48*b*—although the rate of pressure rise is very rapid—the curve definitely slopes away from the vertical. As may be

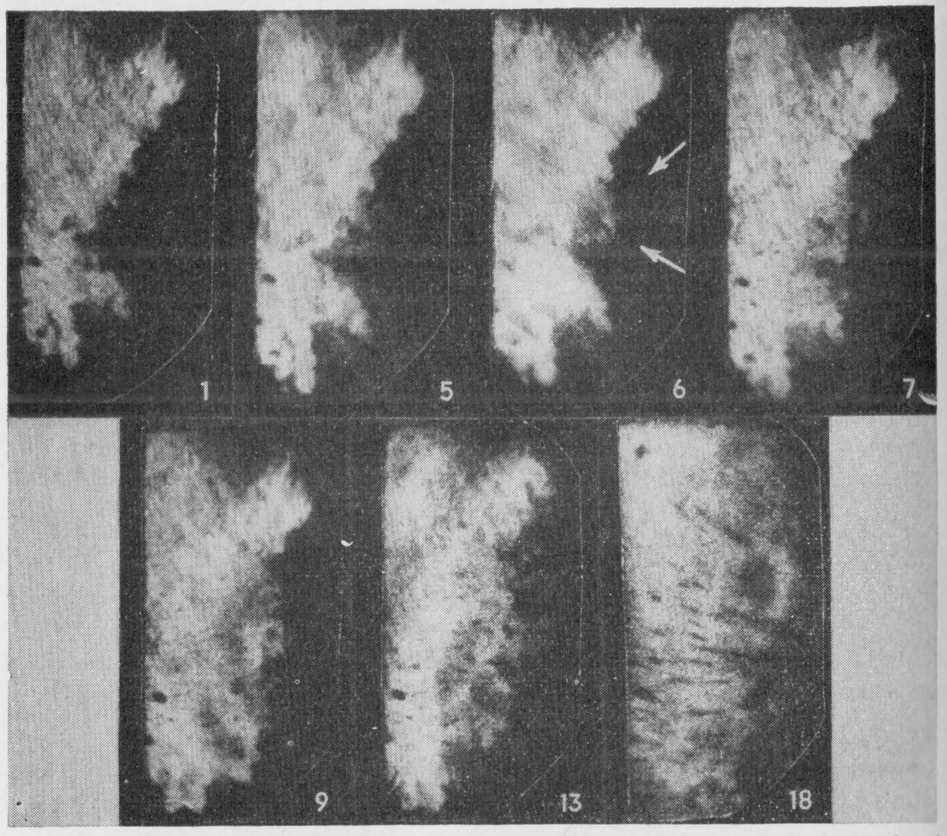
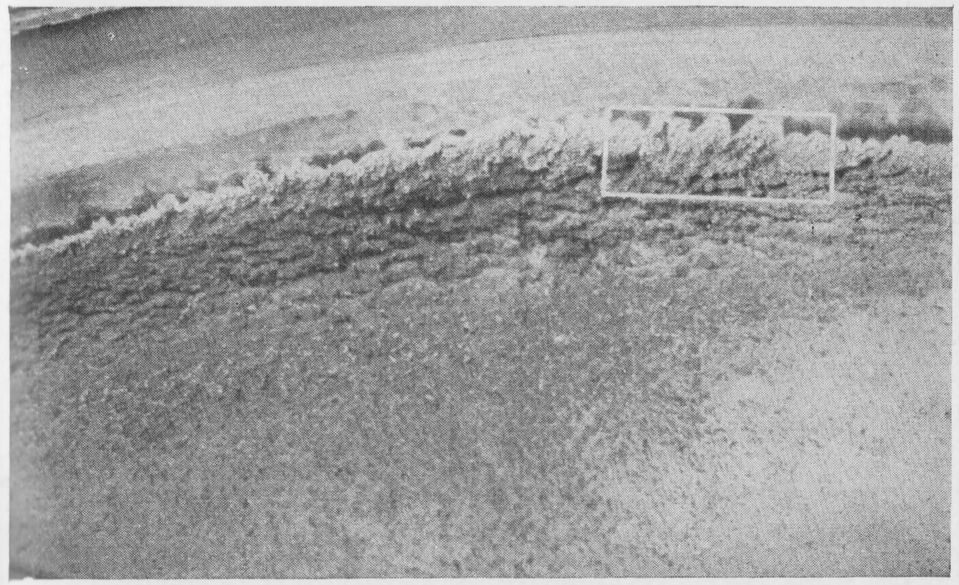


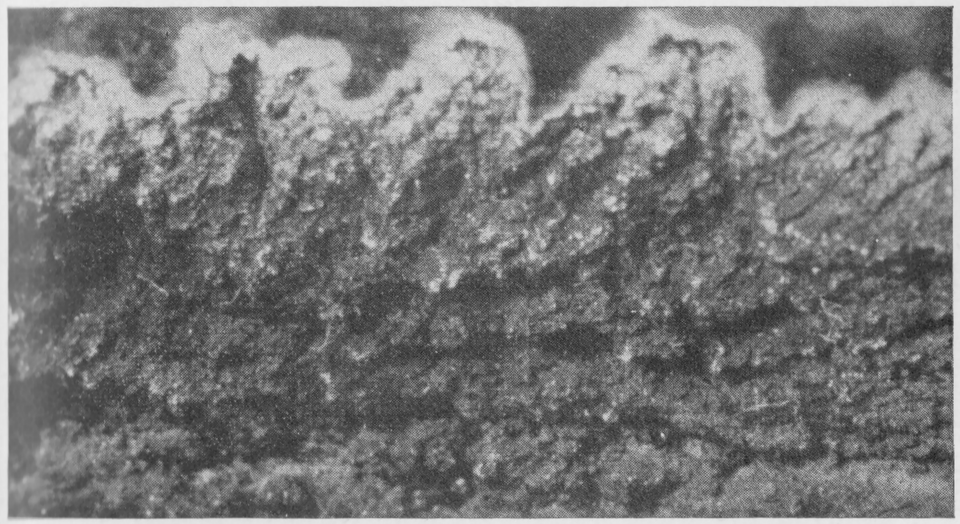
FIG. 50. Motion pictures taken at 200,000 frames per second showing the flame development for the detonation-wave type of knock in a spark-ignition engine. (*NACA Aircraft Engine Research Laboratory.*)

inferred from Fig. 48, there tends to be a difference in the sound from the two types of knocking combustion, the detonation wave giving a high-pitched ringing sound as compared with a thudding knock for plain autoignition.[8] Only high-speed motion pictures or cylinder-pressure diagrams seem to be satisfactory as a means of differentiating between the autoignition- and detonation-wave types of knock, however.

It is important to note that, although detonation gives higher peak pressures than normal combustion and a higher rate of pressure rise



(*a*)

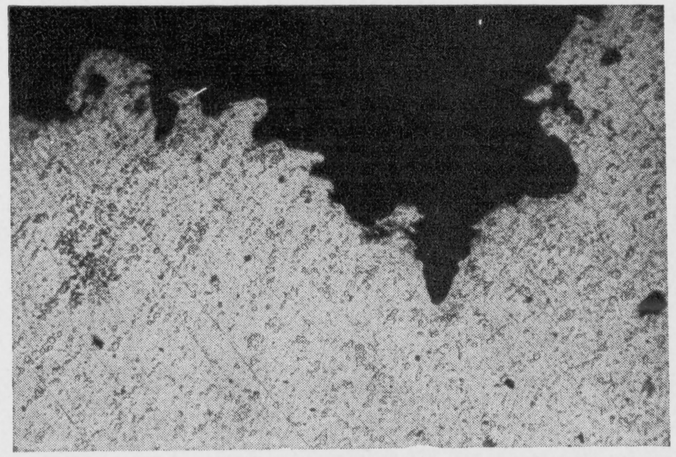


(*b*)

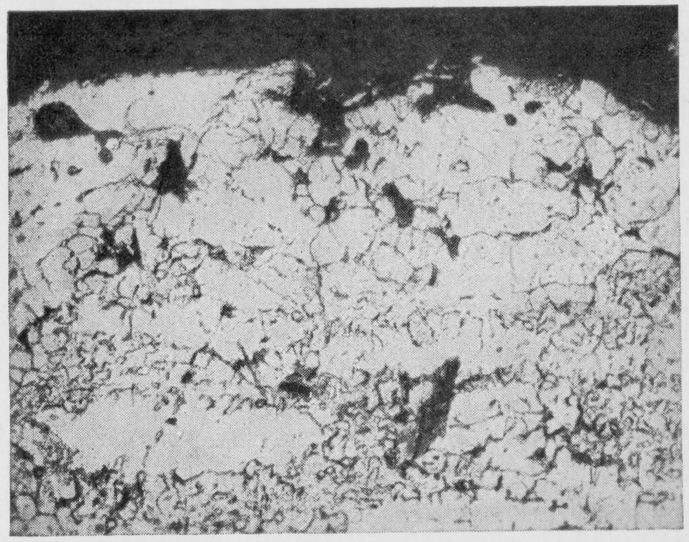
FIG. 51. Photograph showing the appearance of an aluminum surface badly "burned" by detonation. The part shown is an aircraft-engine cylinder head. Magnification of (*a*) about 2× and (*b*) about 8×. (*California Research Corp.*)

for the last portion of the charge to burn, the path of a mean-pressure line on the diagram after combustion has been completed is slightly below the corresponding curve for normal combustion.[8] This seems to be caused mainly by the fact that the detonation wave leaves unburned carbon in the form of smoke so that less heat energy is released, although greater heat losses to the combustion-chamber

walls caused by high local gas velocities are also a factor. The consequent loss in power is shown in the pressure–crank-angle diagrams of Fig. 48*d*, which are for operation both without detonation and with very heavy detonation. The proportion of the charge that detonates



(*a*)



(*b*)

FIG. 52. Photomicrographs of sections through surfaces "burned" by (*a*) detonation and (*b*) by preignition. (*California Research Corp.*)

is normally very much smaller than was the case for Fig. 48*d*. It is important to remember that detonation always occurs late in the combustion process so that only a very small portion of the total charge is normally involved and hence the loss in power is usually too small to measure.

Damage Caused by Detonation and Preignition. Prolonged operation with more than a very small amount of the charge detonating will damage the engine. The damage may take the form of cracks in the cylinder head and/or piston, it may be in the form of scuffed or scored piston rings, pistons, or cylinders as a result of lubrication failure, or it may be in the form of "burning" of the cylinder head or piston. Figure 51 shows a portion of a cylinder head that has been burned by detonation. The actual process is probably not oxidation of the metal but rather appears to be erosion caused by very high local gas velocities in the vicinity of the detonating portion of the charge. While the general appearance of areas burned by detonation is very similar to that of areas burned by preignition, photomicrographs of sections taken through damaged parts of heat-treated aluminum alloy show that the metal just below the burned surface has been annealed in instances in which preignition has occurred, while no loss in heat-treatment can be noted in the structure of the metal adjacent to areas burned by detonation. Figure 52 shows two representative photomicrographs of this sort.

While damage to engine parts is the most serious ill effect of detonation, noise is also an undesirable feature, particularly in passenger-car engines. Detonation will not in itself cause engine stoppage, but it may induce preignition, which in turn is likely to cause backfiring and/or stalling of the engine. Engine stoppage caused by preignition is likely to be a serious matter in aircraft, for it is most likely to occur during the high-power-output conditions of the take-off.

FACTORS AFFECTING KNOCK AND PREIGNITION IN SPARK-IGNITION ENGINES

Either preignition or knock may be induced if the fuel-air charge reaches values of pressure and temperature above a critical set that depend on the particular fuel employed.[13] In general, paraffinic fuels are more inclined to give knocking combustion than preignition, whereas aromatic fuels have the opposite characteristics. While there are again many exceptions, in general it may also be said that an increase in pressure is more likely to result in knock than in preignition. Conversely, but to a lesser degree, an increase in temperature is more likely to induce preignition than knock. The detection of both preignition and knock is confused by the fact that either may under the same conditions induce the other so that difficulties often appear to result from one when actually initiated by the other. Pre-

ignition, for example, may bring combustion to completion earlier in the cycle, *i.e.*, near top center. This would mean a higher pressure in the last part of the charge to burn and thus might induce knock. On the other hand, knocking combustion may cause local overheating and thus a hot spot that will induce preignition. Fortunately, with the types of fuels generally employed in the United States, preignition is only infrequently encountered in water-cooled spark-ignition engines. Thus the two phenomena are usually not concurrent in this type of engine, although they may be in the similar but hotter cylinders of air-cooled engines.

Knock Detection. Engine-test work on factors affecting preignition or knock is complicated by the problem of measuring an intangible and unstable phenomenon. Many types of instrumentation have been devised for the purpose, but the nature of the problem is such that no single set of instrumentation is universally applicable to all engines irrespective of type, size, or manufacturer. In fact, even in a particular engine, the sensitivity of the instrumentation will vary with operating conditions.

An important phase of the problem of knock detection is the determination of knock severity, for such observations are dependent upon both the method of detection and the individual observer. Closely coupled with this matter is the question of the knock intensity that may be tolerated. Noise is often the limiting factor in passenger-car engines, while engine damage is usually the limiting factor with heavy-duty engines. British engineers have an interesting method of rating knock—they state it in terms of the distance in feet from the engine at which a skilled operator can hear an audible knock. A more convenient means is to evaluate the knock intensity from a fixed distance from the engine—say two or three feet depending on the noise level in the room or vehicle. With some experience, and after damaging engines in several tests, an operator can learn to tell the maximum knock intensity that can be maintained without danger of damaging engine parts. Table 11 shows what might be called a *knock spectrum* for one engine model when operated with a fixed rpm, spark advance, compression ratio, and fuel-air ratio. Since the pressure of the unburned portion of the charge before it detonates is the critical consideration and since that pressure varies linearly with the absolute pressure in the intake manifold, the latter has been taken as the reference variable for this chart. Experience with the particular engine for which the chart was prepared has shown that the detonation indications would vary approximately in the manner indicated in the

table as the throttle was opened from the point at which knocking combustion began.

In knock-test work one of the most satisfactory procedures has been found to be that in which the knock intensity is maintained con-

TABLE 11

EFFECT OF INTAKE MANIFOLD PRESSURE ON TYPICAL SYMPTONS OF AND DAMAGE CAUSED BY KNOCK

<table>
<tr><th rowspan="2">Manifold absolute press. - in. Hg</th><th colspan="2">Audible sound</th><th rowspan="2">Appearance of exhaust flames</th><th rowspan="2">Engine behavior</th><th rowspan="2">Part damage from extended operation</th></tr>
<tr><th>Intensity</th><th>Frequency</th></tr>
<tr><td rowspan="6">36
34
32
30
28
26</td><td rowspan="2">Heavy knock</td><td rowspan="5">Continuous</td><td rowspan="2">Heavy smoke</td><td>Back - firing
Abrupt stoppage
Very rough</td><td>Cracked cylinder heads</td></tr>
<tr><td rowspan="2">Somewhat rough</td><td>Cracked pistons
Scored cylinders</td></tr>
<tr><td>Medium knock</td><td>Orange flames
Light smoke</td><td rowspan="3">Smooth</td><td>Lightly burned aluminum pistons
Scuffed rings</td></tr>
<tr><td>Light knock</td><td>Intermittent</td><td>Intermittent bursts of orange color in flames</td><td rowspan="3">No damage</td></tr>
<tr><td>Scarcely perceptible knock</td><td>Occasional</td><td rowspan="2">Clear blue</td></tr>
<tr><td>No knock</td><td>No knock</td><td>No knock</td></tr>
</table>

stant at as low a value as is clearly perceptible. Such conditions are often referred to as *trace-knock* or *borderline-knock* conditions. Such a procedure not only ensures freedom from part damage in all types of engine but also gives a more stable set of conditions; for example, severe knock may induce preignition. For many types of engine, the

human ear has been found to be more satisfactory than any instrument for determining the presence or absence of knock. This is because the ear of a man skilled in knock-test work is very reliable, very sensitive, and free from the service irregularities that invariably accompany electronic instrumentation.

If the noise from the engine valve mechanism is pronounced, it may obscure the audible knock from knocking combustion because of the similarity in the quality of the two types of sound. Again, if the noise level around an engine is high, as is the case if the engine is operated with open exhaust stacks, even a skilled operator may not be able to detect destructive knocking combustion by ear. In such cases one of several types of knock or detonation indicator is essential. Such an indicator may be mounted on or in the cylinder head so that it will respond to the high-frequency vibrations set up by a detonation wave. The output from the electrical type of pickup may be fed through an amplifier to an oscilloscope. The image on the oscilloscope screen will then indicate detonation intensity. Most varieties of this type of pickup are sensitive primarily to the detonation-wave type of knock and are not appreciably affected by the autoignition type of knock unless it is accompanied by a detonation wave (as is virtually always the case with commercial United States fuels). A mechanically more simple form of knock indicator sensitive to all types of knocking combustion is employed on the standard CFR (Cooperative Fuel Research) fuel-test engine used for the octane rating of fuels. A specialized type of instrument, it is not suited to general-purpose testing. It is described in detail in the chapter on fuels.

Preignition Detection. The detection of preignition is a very difficult problem unless some form of cylinder-pressure pickup is employed. The only manifestation to the engine operator of mild preignition is rough or slightly irregular engine operation. Even an experienced observer may be easily misled and feel that the roughness is caused by other factors such as a faulty ignition system. If preignition is occurring at the spark plug of a single-cylinder engine, switching off the ignition for a second or so is a reliable method of detection, for under preignition conditions the engine will *afterfire*, *i.e.*, continue to fire for at least a few cycles. The same procedure may be satisfactory if some other hot spot is responsible. However, difficulty invariably arises if an unstable form of preignition occurs. In such cases, once preignition starts it rapidly grows worse so that in 1 or 2 sec (10 to

50 cycles) it may occur so early during the compression stroke that it will stall the engine. Under such conditions the only satisfactory method of detection is a cylinder-pressure pickup. This may be instrumented to give either a cylinder-pressure–crank-angle diagram or a rate of change of pressure diagram on an oscilloscope.

Detonation-limited IMEP as an Index of the Effect of Engine Variables on Detonation. Because of the difficulty in evaluating knock intensity as well as the wide variation in the tolerance of knocking combustion between various types of engine, it has seemed best to present the effects of the more important operating variables in the terms of the imep at which light intermittent knock would occur, *i.e.*, a "trace-knock" or "borderline-knock" condition. In determining the knock-limited imep for any given set of conditions in this fashion, all operating variables are held constant except the intake-manifold absolute pressure, which is increased by opening the throttle until the trace-knock condition is obtained. It seemed to be an especially suitable method since the imep is directly proportional to the power output, which in turn is really the most important consideration. It happens that an excellent and complete set of data has been determined and presented in this fashion for a supercharged aircraft engine.[14] Similar tests on an automotive engine with regular motor fuel would give similar curves except that the imep would not be as great.

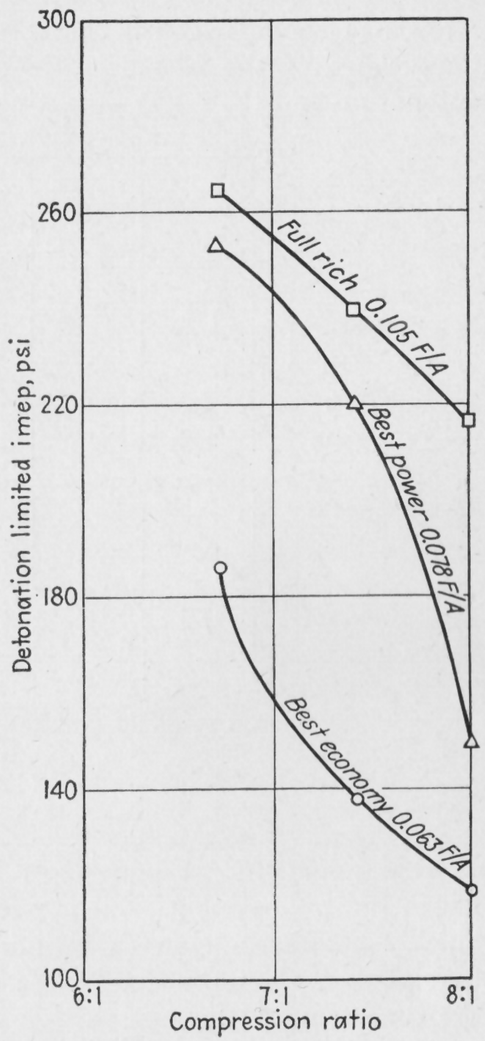


FIG. 53. Effect of compression ratio on knock-limited imep. (*Droegemueller, Hersey, and Kuhrt, Trans. SAE, Vol.* 52, 1944.)

Effect of Compression Ratio on Knock Limit. Figure 53 shows the effect of compression ratio on the detonation-limited imep of an engine for operation with three different fuel-air ratios. As one would expect, the lower compression ratios permit higher imep's because the pressure of the last part of the charge to burn is lower. The difference in shape between the three curves shows something of the importance of the fuel-air ratio on the complex relations between detonation and the many operating variables.

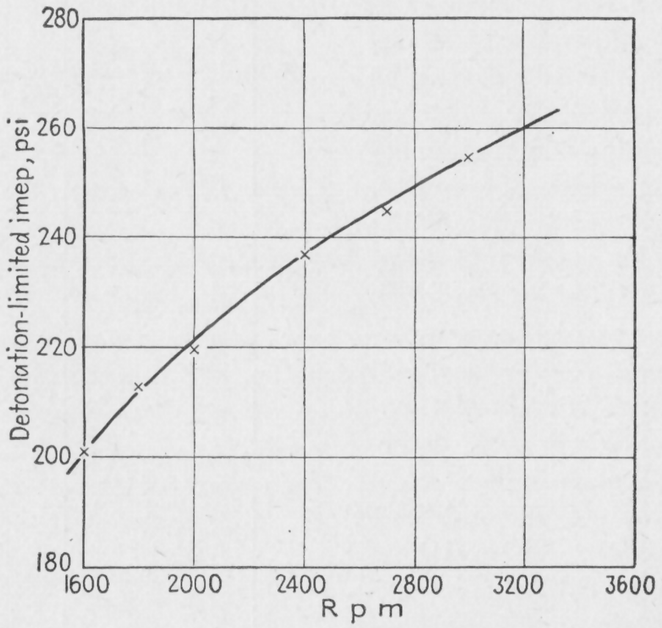


FIG. 54. Effect of rpm on knock-limited imep. (*From unpublished data for a full-scale aircraft engine.*)

Effect of RPM. Engine speed has a considerable influence on the knock limit. As mentioned previously, if the charge can be burned quickly before the slow oxidation reactions that precede knock can take place in the unburned charge, detonation can be avoided. Since the time required for combustion is almost inversely proportional to engine speed, as shown in Fig. 54, an increase in engine rpm leads to a considerable increase in the knock limit.

Effects of Spark Advance on Knock Limit. Detonation-limited curves determined as a function of spark advance are presented in Fig. 55. Since an increase in spark advance means an increase in peak pressure under normal conditions, the knock-limited imep falls off with an increase in spark advance. Note that again the fuel-air ratio has a marked effect on the shape of the curves. While the curves

presented here are typical, the many operating variables do produce considerable differences from one engine to another.

Effects of Fuel-Air Ratio and Manifold Temperature on Knock. The effects of charge temperature in the intake manifold are so interrelated with those of fuel-air ratio that a series of curves such as those in Fig. 56 must be run to describe the characteristics of the fuel properly. Note that as the temperature is increased the knock-limited imep falls off much more rapidly with lean than with rich mixtures unless the mixture is leaned out beyond the stoichiometric value.

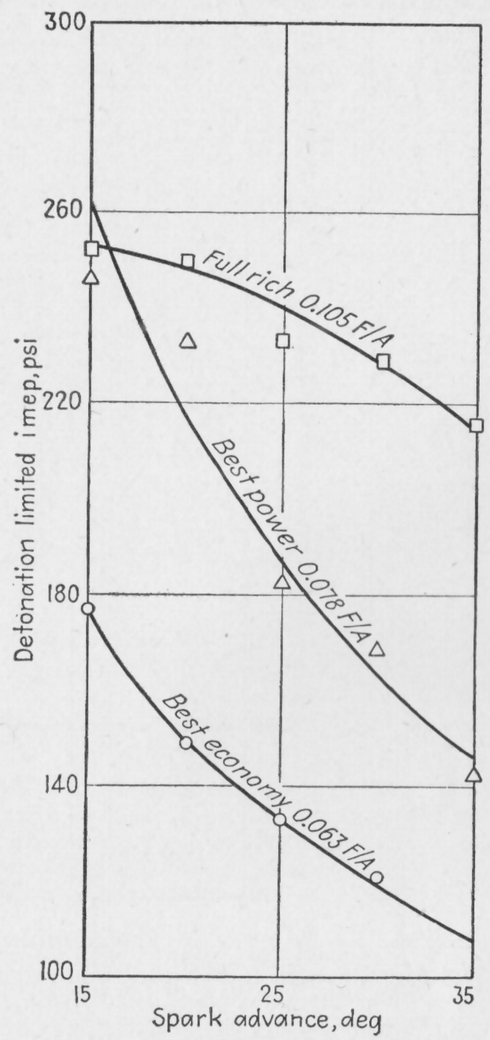


FIG. 55. Effect of spark advance on knock-limited imep. (*Droegemueller, Hersey, and Kuhrt, Trans. SAE, Vol.* 52, 1944.)

Effects of Humidity and Water Injection on Knock. The presence of water vapor in the combustion chamber inhibits knock.[15] At least a part of the effect is probably a result of the reduced temperatures that would accompany the presence of any inert gas having a high specific heat. It seems likely that there is also some chemical effect as well in that the water vapor may inhibit the type of preflame reactions that lead to knocking combustion. Figure 57 shows the effect of water injection into the intake manifold. The effect of the water vapor in humid air would be represented fairly well by the same curve. Note that the curve levels off at the higher water-air ratios, where appreciable amounts of water are passed into the cylinder unvaporized. When this happens in multicylinder engines, some cylinders will not get their proper share of water and knock will occur. Perhaps the most important effect of water injection is that it acts to inhibit knock by reducing the intake manifold

temperature as it evaporates. The great effectiveness of this temperature drop, which may be as much as 150°F, may be deduced from both Figs. 56 and 57.

Effect of Cylinder-head-metal Temperature on Knock. Figure 58 shows how rapidly the knock-limited imep falls off as the manifold

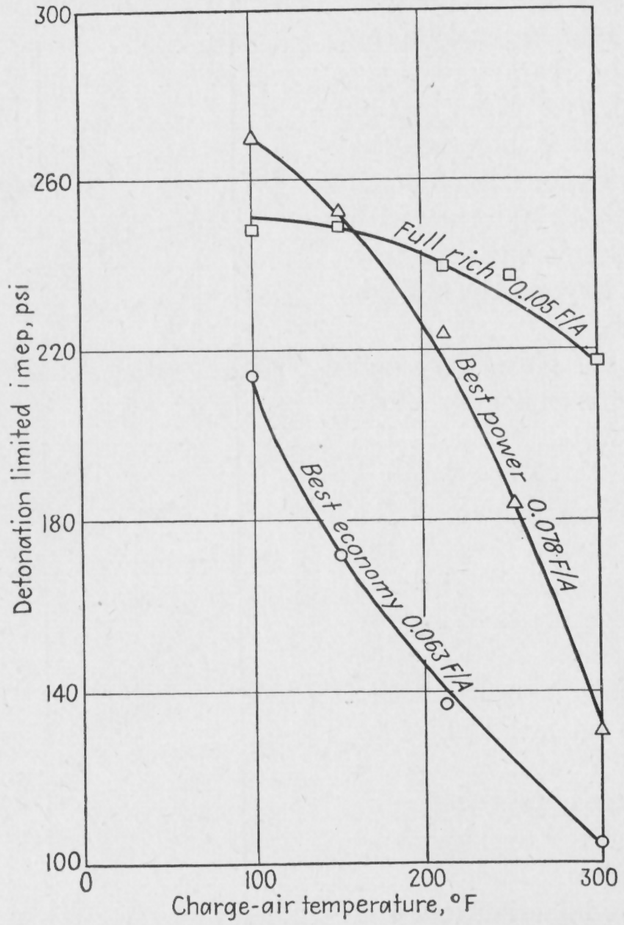


FIG. 56. Effect of fuel-air ratio and intake manifold temperature on knock-limited imep. (*Droegemueller, Hersey, and Kuhrt, Trans. SAE, Vol.* 52, 1944.)

temperature is increased and the relatively much smaller loss resulting from an increase in cylinder-head temperature. These curves show very forcibly an important advantage that the liquid-cooled engine has over an air-cooled engine, which must operate with higher cylinder-head-metal temperatures.

FLAME MOVEMENT AND PRESSURE DEVELOPMENT IN DIESEL ENGINES

The combustion processes in spark-ignition and compression-ignition engines have many points in common as well as many inherent

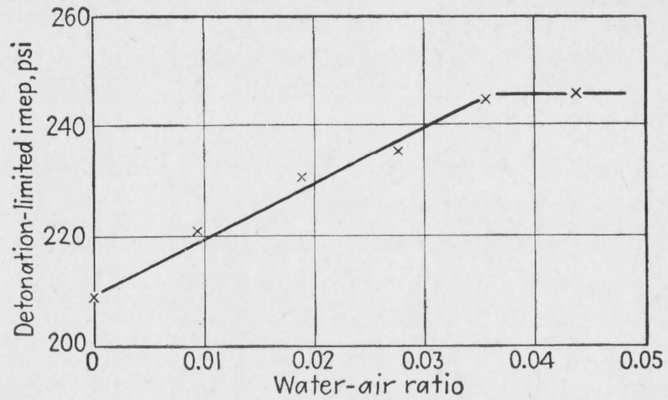


FIG. 57. Effect of water injection into the intake manifold on knock-limited imep. (*From unpublished data for a full-scale aircraft engine.*)

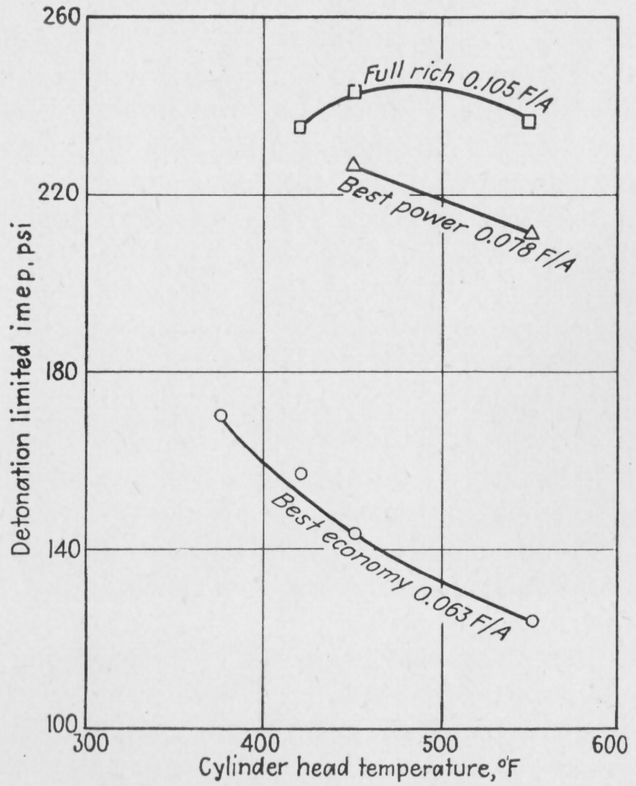


FIG. 58. Effect of cylinder-head temperature on knock-limited imep. (*Droegemueller, Hersey, and Kuhrt, Trans. SAE, Vol.* 52, 1944.)

differences. The background of fuel composition and combustion chemistry presented in the first portion of the chapter is equally applicable to both types. The principal differences in practice arise from the fact that diesel combustion takes place in a nonhomogeneous charge in which liquid fuel, fuel vapor, mixtures of fuel vapor and air, and air practically free of fuel vapor coexist in the combustion cham-

ber during most of the combustion process. The degree to which the combustion chamber and fuel-injection system tend to give a homogeneous strata of combustible fuel-air mixture in the short time available determines to a large extent the completeness of the combustion reactions and hence the degree to which the performance of a diesel engine approaches the optimum value.

Injection and Ignition of Fuel. The high-speed motion pictures of Fig. 59 show the injection, ignition, and combustion of the fuel charge in a diesel engine.[16] The injection nozzle is designed to give a very finely atomized spray that is distributed as uniformly as possible throughout the combustion chamber. Note in the pictures that a considerable amount of fuel is injected into the cylinder before the first portion injected tends to ignite. As soon as ignition occurs, a flame spreads outward from the first portion of the fuel vapor to ignite in somewhat the same manner as in spark-ignition engines. In the diesel, however, other flame fronts begin at random times and locations in the combustion chamber, and the rate of flame propagation outward from them is highly variable because of charge stratification into locally rich or lean regions. This factor also causes combustion to continue in overrich sectors for a long time after the passage of the flame front. Note, too, in Fig. 59 that fuel never reaches some portions of the air charge so that the oxygen in those portions cannot be utilized for combustion. This is the reason why air turbulence in diesel combustion chambers is so important, for it greatly influences the degree of mixing of the fuel and air and hence determines the amount of fuel that can be burned effectively in a given combustion chamber. In some diesel engines a *precombustion chamber* is provided in the form of a bulb connected to the cylinder by a restricted passage. A section through such an arrangement may be seen in Fig. 111 (page 185). The fuel injected into the precombustion chamber burns with the air there. The large rise in pressure accompanying combustion forces the hot gases containing a considerable amount of unburned fuel vapor from the precombustion chamber out into the air in the cylinder proper at a high velocity. This action brings about a high degree of turbulence with intimate mixing of the unburned fuel and air to give excellent conditions for combustion. The principal disadvantages of such an arrangement are the power loss arising from the high gas turbulence, the higher rate of heat transfer to the walls, and the structurally weaker cylinder head.

Effect of Fuel-Air Ratio. The useful range of fuel-air ratios in a diesel is limited by excessive smoke in the exhaust and carbon forma-

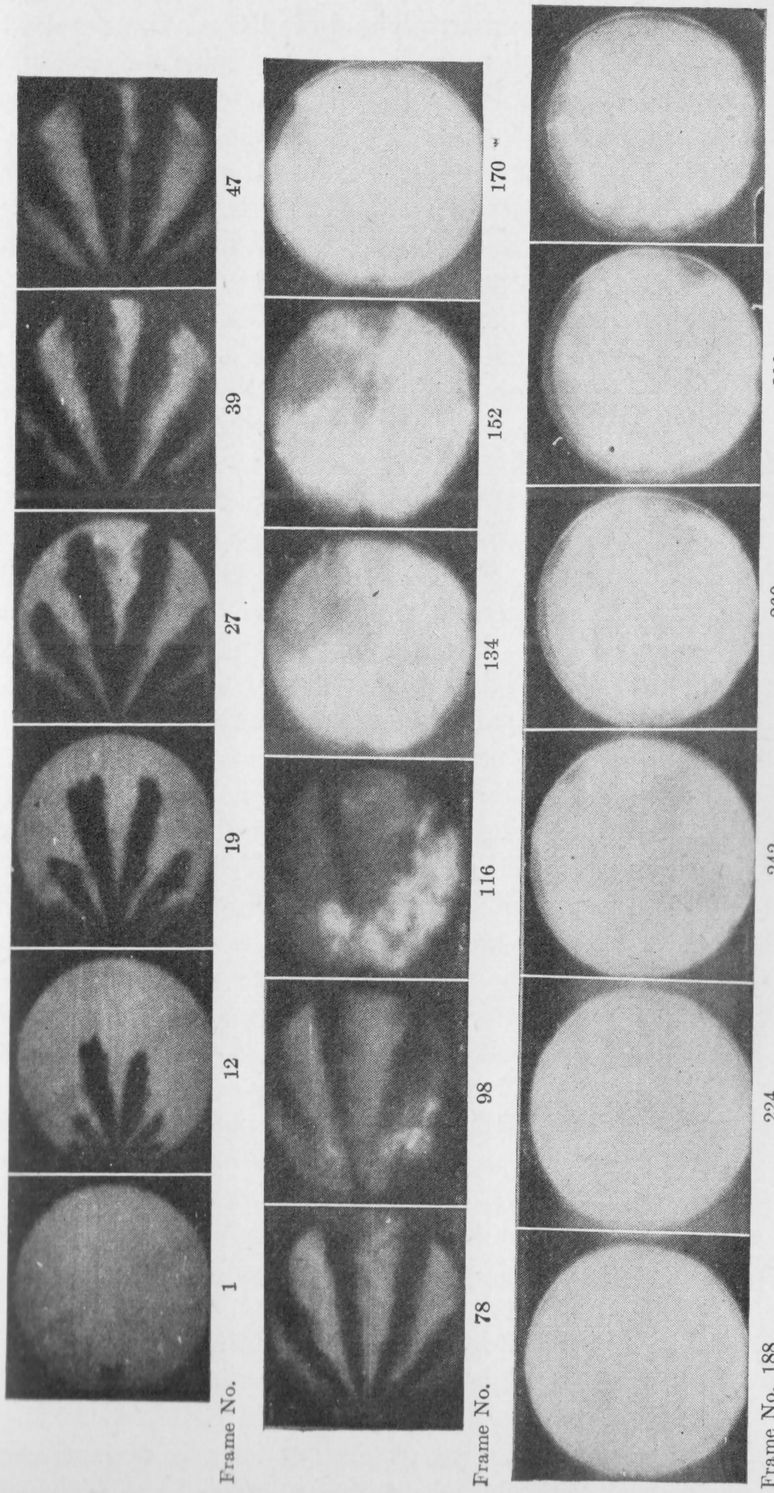


FIG. 59. Motion pictures taken at 40,000 frames per second showing injection and combustion of the fuel for normal diesel-engine combustion conditions. (*NACA Aircraft Engine Research Laboratory.*)

tion in the combustion chamber. The breadth of the smoke-free range is determined by the characteristics of the injection nozzle, the shape of the combustion chamber, and the characteristics of the fuel.[17] Smoke-free operation at high loads (*i.e.*, high fuel-air ratios) can be obtained up to the point where some portion of the combustion chamber is supplied with an excess of fuel so that the mixture there is considerably richer than the chemically correct value. Soot is formed like that from a gas flame with an inadequate admixture of air. This gives a smoky exhaust. Smoke-free operation at light loads (*i.e.*, low fuel-air ratios) can be obtained unless the speed and load are decreased to the point where incomplete combustion results from either low injection pressures and poor atomization of the fuel or low combustion-chamber temperatures, or both. The problem is usually not serious except in the smaller bore engines, where the large ratio of surface to volume in the combustion chamber causes a relatively high heat loss to the walls. This tends to chill the burning mixture before the reactions have proceeded to completion and thus is inclined to give a hazy exhaust under idling conditions.[17] As will be shown later on, an increase in the ignition quality of the fuel will help to increase the breadth of the smoke-free range at both high and low loads. While improvements in the combustion chamber and injection equipment can be made to extend the range "from smoke to smoke," poor combustion and smoke can be expected if a diesel engine is to be operated at too high a fuel-flow rate at any speed and are likely at too low a fuel-flow rate at low idling speeds. Since the richest mixtures practicable in a diesel are leaner than the chemically correct value, the degree to which a combustion chamber is able to utilize all the air available is a good measure of its performance. This has led to the use of the term *per cent excess air* instead of fuel-air ratio in diesel work. The smaller the percentage of excess air for a given smoke density at high loads, the better the performance of the combustion chamber.

Effect of Fuel Ignitability. The lower the spontaneous ignition temperature of the fuel, the earlier in the injection process will combustion begin. That is, if the air temperature in the combustion chamber is only slightly greater than the ignition temperature of the fuel, it will take a relatively long time for a flame to get started. Thus, with a given set of engine operating conditions and hence a particular charge temperature, the time interval between the beginning of injection and the beginning of combustion will depend on the ignitability of the fuel. This time interval is termed the *ignition lag*

and is often referred to in diesel-engine work. The fuel ignitability is usually given in terms of the *cetane number*—the higher the cetane number, the higher the ignitability of the fuel and the smaller the ignition lag.

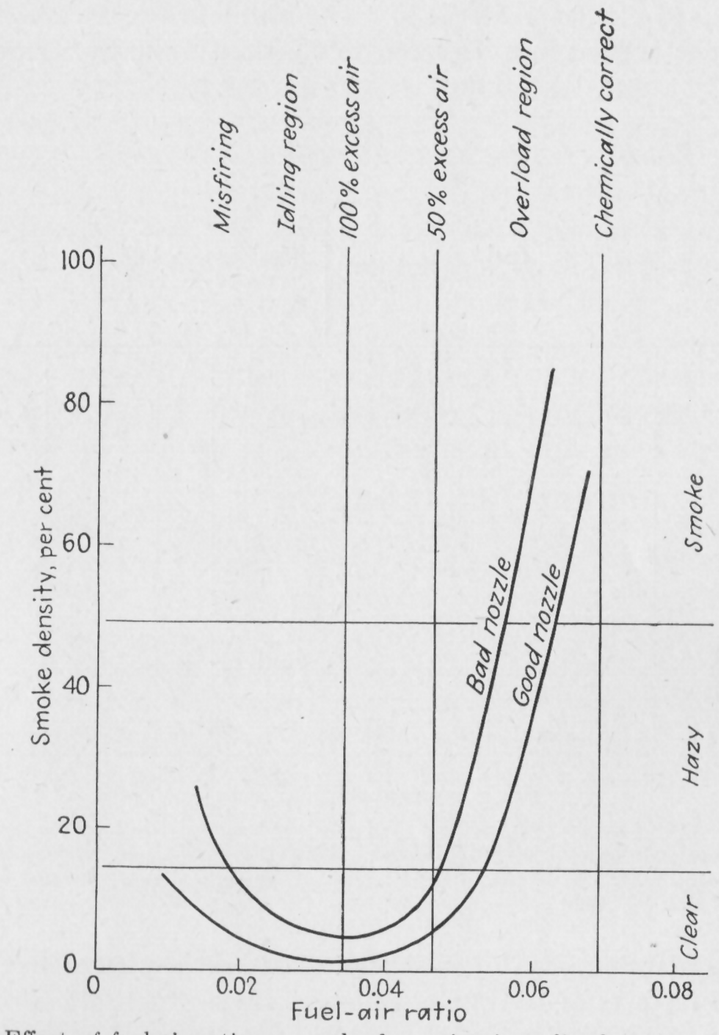


FIG. 60. Effect of fuel-air ratio on smoke formation in a diesel engine. (*Schweitzer, SAE Quarterly Trans., Vol. 1, No. 3, 1947.*)

Effect of Engine Speed. It was pointed out in connection with spark-ignition-engine combustion that the turbulence in the combustion chamber is directly proportional to engine speed. As in the spark-ignition engine, the rate of combustion in a diesel is so greatly dependent on turbulence that the time required for combustion is roughly also inversely proportional to rpm. That is, the number of degrees of crank rotation required for combustion is practically inde-

pendent of speed. This effect is shown in Fig. 61 for an engine operating at full load with four distinct fuels having different cetane numbers.[18] Four types of heavy lines are used in the diagram, one for each of these fuels, as indicated in the legend. The basic coordinate is time plotted along the abscissa. The ordinate is rpm, while a network of light curved lines to indicate constant values of crank angle

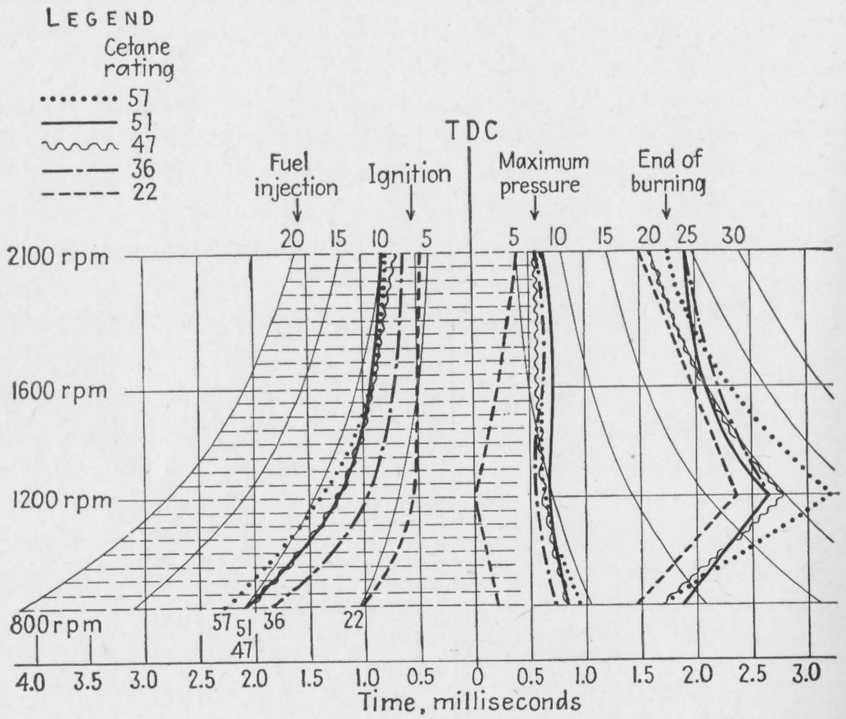


FIG. 61. Effect of rpm and fuel cetane rating on the crank position at the beginning of ignition, at the point of maximum pressure, and at the end of combustion in a diesel engine operating with a constant injection advance angle. (*Shoemaker and Gadebusch, Trans. SAE, Vol.* 54, 1946.)

before and after top center is superimposed on the rpm-time coordinates. Three sets of heavy lines plotted from experimental data are shown, viz., one set for the beginning of combustion (ignition), one set for the maximum pressure, and one set for the end of the principal part of the burning process. The injection advance was maintained at a constant value of 20 deg for all conditions. Note how little effect engine speed had on the number of degrees required for each stage of the combustion process, particularly the ignition lag. Note, too, that the point at which the burning process ended varied little with the ignitability of the fuel but that the ignition lag, or the point at which combustion began, varied widely with the fuel ignitability.

Thus the rate of combustion was greatest for the least ignitable fuel once the flame got started. This is reasonable if one remembers that, the longer the time available, the further the progress of the preflame reactions and hence the more rapid the combustion once the flame does get under way.

Ignition Lag and Knocking Combustion. The most critical single problem in diesel combustion is the time that elapses between the beginning of injection and the beginning of combustion. In the pictures shown in Fig. 59 this period amounted to 78/40,000 sec, or 5.8 deg of crank rotation, a normal value. If the ignition lag is too great, however, combustion will take place at a very rapid rate when the flame does finally get started. Frames from a high-speed motion picture of this very rapid combustion, or *diesel knock*, are shown in Fig. 62.[16] In examining these pictures it is evident that the fuel spray vaporizes and forms a homogeneous mixture with no sign of combustion up to frame 207. The small light area in the upper right-hand corner of that frame grows rapidly in subsequent frames until practically all the charge appears to have been burned by frame 220. Thus virtually all the fuel charge was burned in about 13/40,000 sec, or in about one-sixth of the time required for the normal combustion process shown in Fig. 59. The phenomenon seems to be closely related to knock in a spark-ignition engine. That is, certain preflame reactions occur between unburned fuel vapor and oxygen of which aldehydes are prominent products. The portions of the mixture in which these reactions have taken place may burn almost instantaneously to give an exceptionally rapid increase in pressure, which may in turn produce an audible knock much like that encountered with the autoignition type of knocking combustion in spark-ignition engines. Instances of the detonation-wave type of knock in the diesel have also been reported. As in spark-ignition engines, damage to diesel-engine parts will result from prolonged operation under conditions of severe knocking combustion.

Comparison of Knocking Combustion in Spark-ignition and Diesel Engines. In examining the points of similarity and difference in knocking combustion between spark-ignition and diesel engines a number of points become evident. First, a fuel that ignites spontaneously at relatively low temperatures will be inclined to knock in a spark-ignition cylinder. The same fuel, on the other hand, will prevent knock in a diesel, for it will tend to ignite in the initial stages of the injection process and hence will not give an opportunity for an appreciably large amount of unburned fuel vapor and air to accumu-

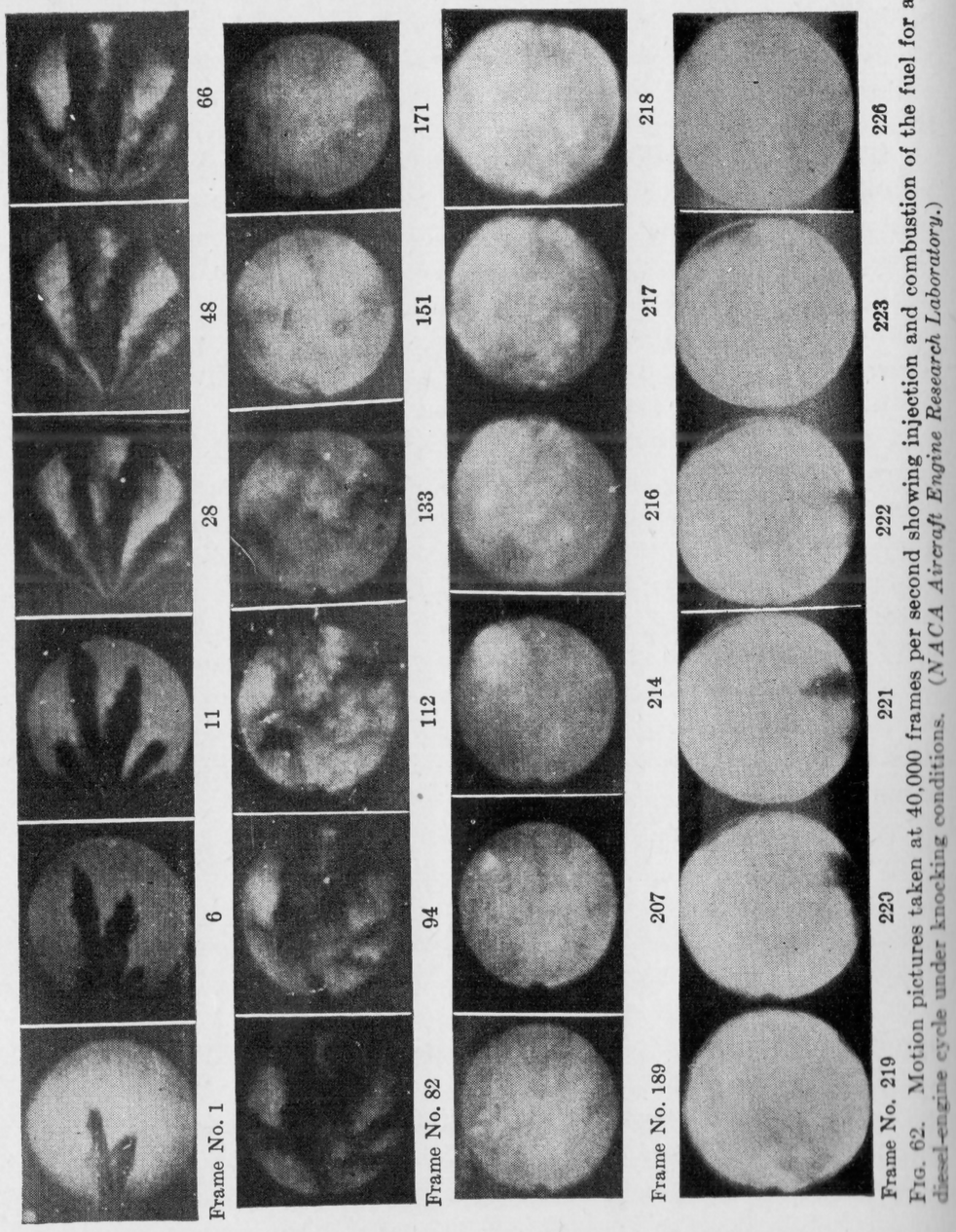


FIG. 62. Motion pictures taken at 40,000 frames per second showing injection and combustion of the fuel for a diesel-engine cycle under knocking conditions. (*NACA Aircraft Engine Research Laboratory.*)

late, *i.e.*, will not provide the necessary prerequisite for knocking combustion in the diesel. Thus the type of fuel desired for one type of engine is just the opposite from that desired for the other. This is the reason why the long straight-chain paraffins that ignite so readily are both so desirable in a diesel fuel and so undesirable in a gasoline.

In reexamining the conditions preceding knocking combustion, it may be noted that the knocking mixture in a spark-ignition engine is almost perfectly homogeneous and hence is affected uniformly throughout by the preflame reactions which bring about knock. In contrast to this, the fuel-spray zone in a diesel cylinder is a heterogeneous mixture of liquid fuel droplets, fuel vapor, and air so that it cannot be uniformly affected by preflame reactions, and hence knock is usually not possible in the same sense as in a spark-ignition engine.

Ignition Lag as an Indication of Diesel Knock. While knock in a diesel is less clearly defined than in a spark-ignition engine, it is evident that the amount of fuel-vapor–air mixture which would be inclined to knock in a diesel increases with the quantity of fuel injected before inflammation begins. This, in turn, is a function of the lag between the beginning of injection and the beginning of inflammation. *Therefore, the tendency of a diesel fuel to knock is directly related to ignition lag.*

FACTORS AFFECTING IGNITION LAG IN DIESEL ENGINES

Measurement of Ignition Lag. As was pointed out above, ignition lag is closely related to the rate of cylinder pressure rise. Since the ignition lag is much easier to measure than either the rate of pressure rise or that intangible quantity "knock intensity," it is a widely used parameter in diesel-engine work. Ignition lag may be determined by any of several methods. Flame pictures such as those in Fig. 59 provide the most accurate but also the least convenient means. Pressure–crank-angle diagrams may be employed for the purpose by determining the angular difference between the beginning of injection and the point at which the curve departs from the diagram for a simple compression cycle without ignition. The simplest procedure, however, is to determine the degrees of crank rotation between the beginning of injection and the point at which the cylinder pressure exceeds the compression pressure (*i.e.*, the peak cylinder pressure for a simple nonfiring cycle). Since injection-advance angles are normally less than 30 deg, the error involved in the latter procedure is quite small.

Effect of Injection Timing. The effects of varying the injection advance (*i.e.*, the timing of the beginning of injection) on the com-

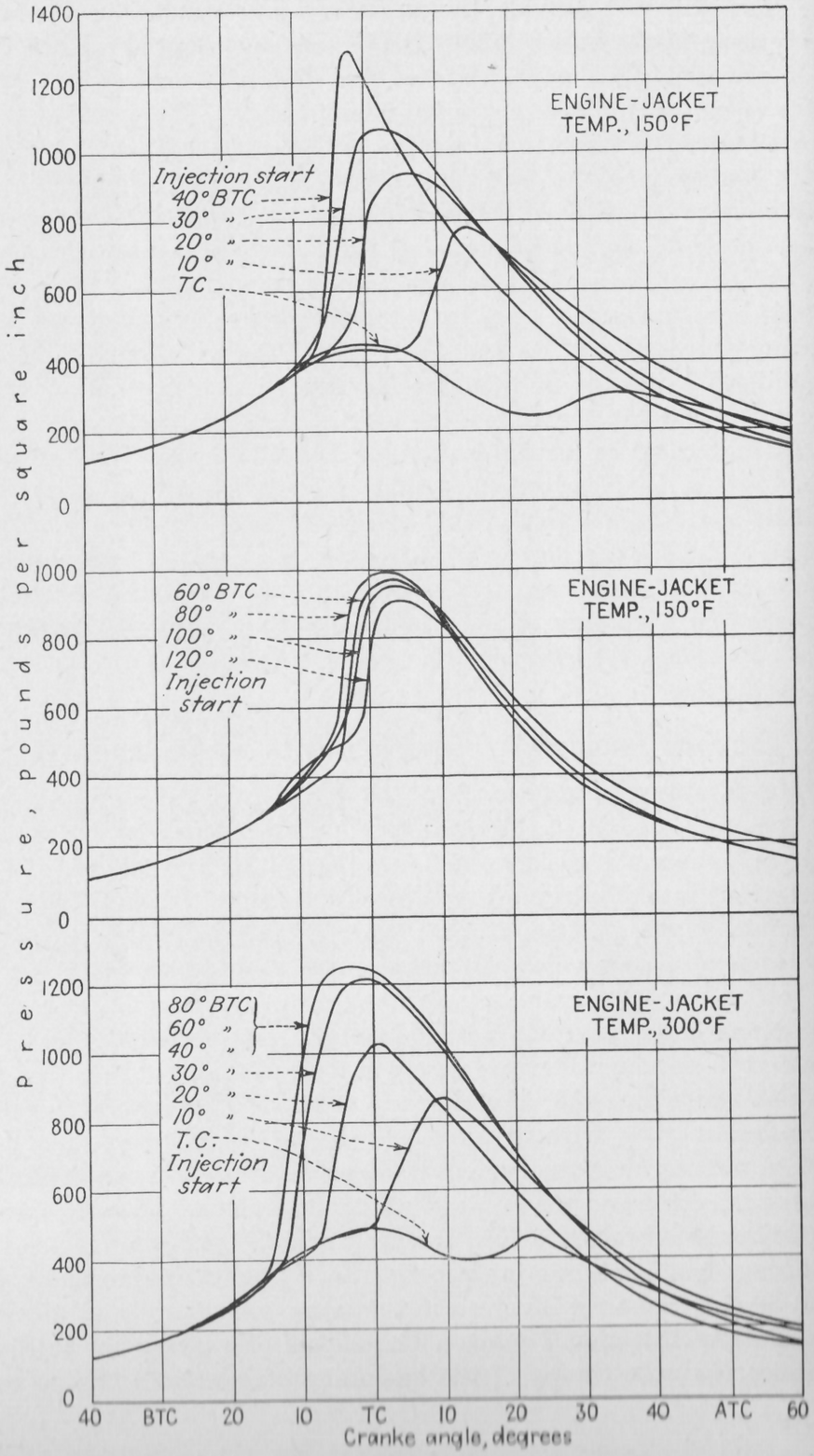


FIG. 63. Effect of injection-advance angle and cooling-jacket temperature on the pressure-crank angle diagrams for a diesel engine. (*Rothrock and Waldron, NACA Tech. Rept.* 525, 1935.)

bustion rate and hence the rate of pressure development in a diesel cylinder are somewhat similar to but more complex than the corresponding effects of spark timing in a spark-ignition engine.[19] Figure 63 shows a set of pressure–crank-angle diagrams for a series of injection-advance settings.[20] In a spark-ignition engine, variations in spark timing have a marked effect on the appearance of the indicator diagram but have little effect on the combustion rate as indicated by

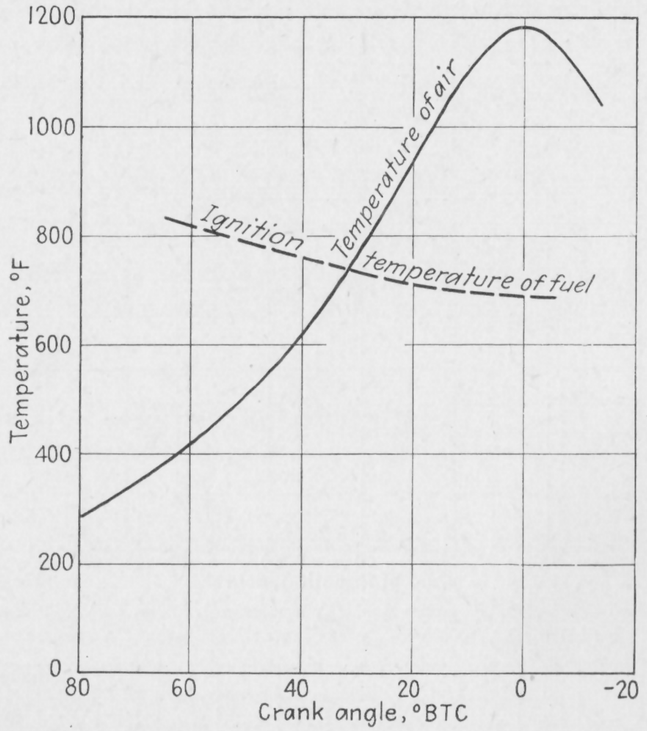


FIG. 64. Effect of crank angle on the spontaneous ignition temperature of a fuel and the temperature of the air in a diesel cylinder. Compression ratio = 14.0:1 and initial air temperature = 112°F.

the maximum slope, or rate of pressure rise. On the other hand, as shown by Fig. 63, in a diesel engine the injection advance affects the point at which combustion begins only to a small extent, but it greatly affects the rate of combustion.

The reasons for and the nature of the effect of injection timing on ignition lag are shown nicely in Fig. 64, which shows cylinder-charge temperature vs. crank angle. The greater the amount by which the temperature of the air charge exceeds the ignition temperature of the fuel when injection begins, the smaller the ignition lag. Thus Fig. 64 shows how advantageous it is to have the injection take place close to the top-center position of the piston. The importance of this con-

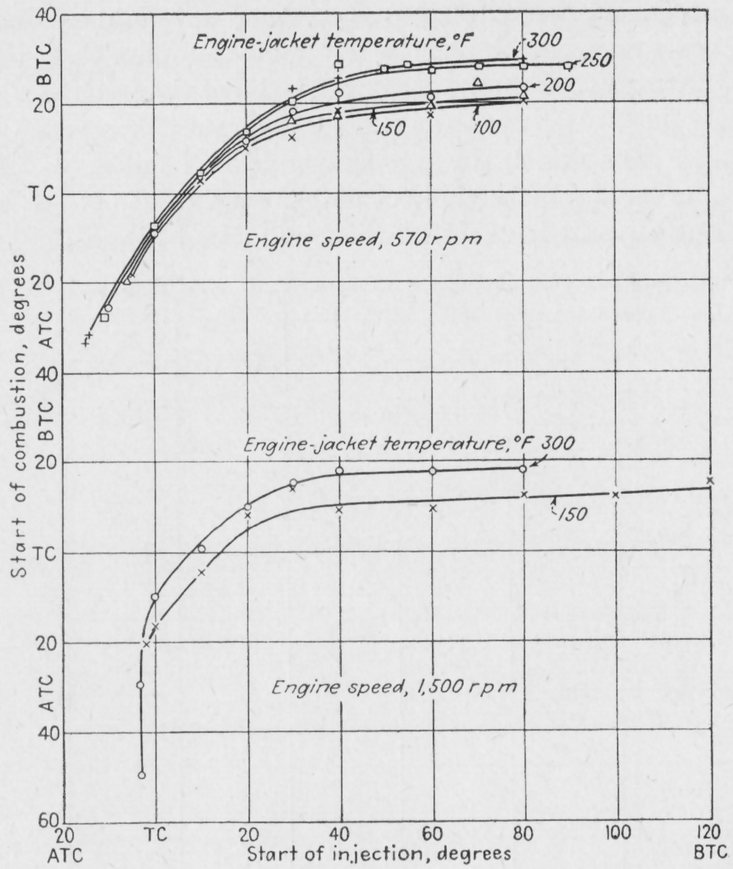


FIG. 65. Effect of injection-advance angle on the crank angle at which combustion begins and on the ignition lag at different jacket temperatures; engine speed 570 rpm. (*Rothrock, NACA Tech. Rept.* 401, 1932.)

sideration is shown in Fig. 65*a*, in which it is clear that increasing the injection advance beyond a certain point has no influence on the point at which inflammation begins.[21] Of course, the rate of pressure rise once inflammation does begin tends to increase rapidly with ignition lag. Figure 65*b* shows the data of Fig. 65*a* replotted to show the effect of injection advance on ignition lag directly.

Compression Ratio. The effects of compression ratio on ignition lag may be deduced from Fig. 66. Note that the increase in pressure with compression ratio brings about some reduction in the ignition temperature. In practice, it has been found that compression temperatures

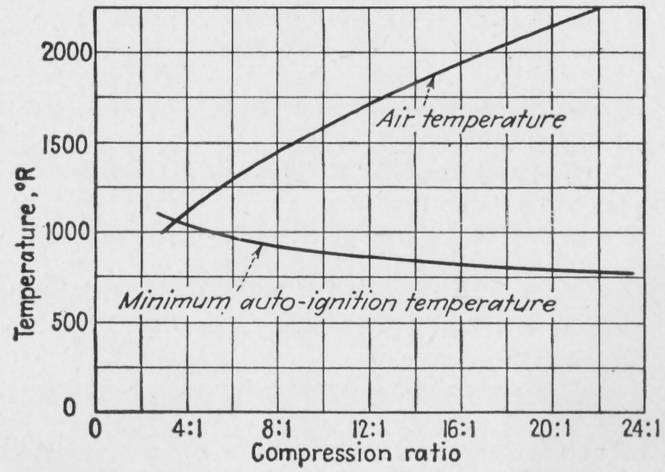


FIG. 66. Effect of compression ratio on air temperature at top center and on minimum autoignition temperature. (*Rothrock, NACA Tech. Rept.* 401, 1932.)

are not high enough for good ignition conditions for starting or light-load operation unless compression ratios in excess of about 14:1 are employed. This is particularly true under starting or low-load conditions, where the cylinder walls cool the air charge considerably during compression. Since peak cylinder pressures increase rapidly with compression ratio, most diesel engines use the lowest compression ratio that gives good ignition conditions.

Inlet Air Temperature. Since the temperature of the air in the cylinder during compression is directly proportional to that of the air entering the cylinder, that factor is also important. The effect of inlet air temperature on ignition lag may be deduced from Fig. 64.

Cylinder Temperature. Cylinder temperature has much the same effect on ignition lag as inlet air temperature. The effects of a rather wide range of temperatures of the coolant in the cylinder cooling jackets is shown in both Figs. 63 and 65. The effects of cylinder-

jacket temperature are greatest at light loads, where cold walls tend to have a proportionally greater effect on the charge temperature toward the end of the compression stroke. As will be discussed in Chap. XI, Cooling, considerations other than those of combustion act to limit engine-cooling water temperatures to 150 to 200°F in most cases. These considerations include service limitations, the characteristics of the coolant, etc., so that temperatures in practice are usually lower than the optimum value from the combustion standpoint.

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Problems

1. Determine the chemically correct fuel-air ratios for the following fuels, and the composition of the products of combustion both by volume and in per cent by weight:

Carbon	Methane
Hydrogen	Octene, C_8H_{16}
Carbon monoxide	Benzene, C_6H_6

2. Calculate the heat released per pound of air for the mixtures of Prob. 1.

3. Find the chemically correct fuel-air ratio for a gasoline having a hydrogen-carbon ratio of 0.174.

4. Estimate the temperature of the products of combustion for constant-pressure combustion for each of the fuels in Prob. 1. Assume standard atmospheric initial conditions, and use the specific heats given in Fig. 34. Neglect the effects of dissociation.

5. Compute the chemical energy of combustion available from the fuel in Btu per pound of air for octene-air mixtures having fuel-air ratios of 0.0543, 0.0611, and 0.0677 if the mixture in each case contains 1 lb of air. Take the heating value of octene as 19,190 Btu/lb.

6. Using charts, determine the temperature of the products of combustion of octene and air under constant-volume conditions if the initial temperature is 1000°R and the initial pressure 125 psia. Determine values for fuel-air ratios of 0.0543, 0.0611, 0.0677, 0.0746, and 0.0814, and plot a curve of flame-temperature vs. fuel-air ratio.

7. List three important differences between normal combustion in a diesel and normal combustion in a spark-ignition engine.

8. Distinguish between normal combustion, preignition, and knocking combustion in spark-ignition engines, using sketches of the pressure-time (P-T) diagram.

9. List five factors that, if increased, will tend to cause knock in a spark-ignition engine.

List four other factors that, if increased, will tend to inhibit knock.

10. Distinguish between knock in a spark-ignition engine and knock in a diesel, using P-T diagrams.

11. List four factors that, if increased, will tend to inhibit knock in a diesel.

12. List four possible ill effects of knock in an aircraft engine.

13. Define the following terms:

Chemically correct mixture
Stochiometric fuel-air ratio
Lean combustibility limit
Ignition temperature
Rich mixtures
Knock-limited imep
Trace-knock conditions
Hydrogen-carbon ratio
Spark advance
Ignition lag

CHAPTER V

CYCLE ANALYSIS

A clear conception of the relationships between the ideal and the corresponding actual cycles is essential to a good understanding of engine performance. While there are many factors that tend to cause differences between the two, their collective effects are not too great in a well-designed engine in good running condition if the spark or injection timing, the valve timing, and other factors are properly adjusted. This chapter is concerned with the factors responsible for and the magnitude of the deviations of the actual cycle from its idealized counterpart.

COMPARISON OF IDEAL AND ACTUAL CYCLES

Instruments for Taking Indicator Diagrams. Pressure-volume diagrams for comparison with an ideal cycle can be obtained from a test engine through the use of a cylinder-pressure indicator. The type of indicator employed in steam-engine work is not satisfactory for use in high-speed internal-combustion engines because the inertia of the moving parts is too great. While many varieties of high-speed cylinder-pressure indicator have been employed,[1] there are three main types in general use. These are the balanced-pressure valve (or diaphragm), the piezoelectric, and the capacitance type of pickup. All three overcome the problem of inertia by depending on one or more electric circuits to relay a signal from a pickup mounted on the engine cylinder to the recording instrument and are so designed that the effects of inertia within the pickup on the indicator diagram are negligible at even the highest engine speeds.

The balanced-pressure valve or diaphragm makes what is probably the simplest of the three types of cylinder-pressure pickup. In some models a source of high gas pressure such as a nitrogen cylinder is connected to one side of a small, floating, double-seated valve in the body of the pickup (which is mounted in the cylinder head). If the cylinder pressure exceeds the pressure maintained behind the valve, the valve will rest on its outer seat. Conversely, if the cylinder pressure is less than the pressure maintained behind the valve, it will rest on its inner seat. A steel diaphragm is often used in place of the valve.

In either case an electric circuit is arranged so that it will be interrupted when the valve (or diaphragm) shifts from one seat to the other. The crank angle at which the shift occurs can then be determined in a number of ways. In one method the electric impulse from the shift is used to control a strobotach, and illumination from the flashing neon light of the strobotach is employed to read graduations on the engine flywheel. The pressure of the air or nitrogen behind the sensitive element in the pickup is manually controlled with a valve in the supply line, and its value read from a simple Bourdon-tube pressure gauge. (The data for the indicator diagrams in Figs. 67 and 85 were obtained in this way by reading the crank angle at each of a series of constant pressures maintained behind a balanced-pressure valve.) A more elaborate modification of this basic type makes use of a drum around which is wound a strip of thin paper. The drum is rotated by the engine at crankshaft speed to give a measure of crank angle. An electrode is mounted on a piston, cylinder, and spring arrangement connected to the balance pressure behind the pickup so that the axial position of the electrode depends on the balance pressure. The electrical impulse from the pickup is amplified to give a spark that jumps a gap between an electrode and the drum and perforates the paper. The pressure–crank-angle diagram is thus formed directly.

The sensitive element in most piezoelectric pickups is a quartz crystal. The crystal is usually mounted between a diaphragm exposed to the cylinder pressure and a rigid support in the pickup. The electrical impedance of the crystal is directly proportional to the load imposed on the diaphragm by the cylinder pressure. The pickup output is of the order of 2 or 3 mv per 100 psi pressure. This electrical output can be amplified with a special high-linearity amplifier and the signal fed to the vertical plates of an oscilloscope. The horizontal sweep of the oscilloscope can be synchronized with the engine so that the sweep frequency is either engine speed or some multiple of engine speed. The pressure–crank-angle diagrams in Fig. 48 were obtained in this way. This type of pickup faithfully records pressure fluctuations of even so high a frequency as those of detonation (around 30,000 cps). The principal disadvantage of this method of taking indicator diagrams is that the velocity of the horizontal sweep of a cathode-ray oscilloscope is not constant but falls off as it moves from left to right so that a given horizontal distance on the screen represents a greater amount of time at the right than at the left of the diagram. Because of this it is difficult to relate pressure to crank angle and hence to cylinder volume.

The third type of pickup depends on a variable electrical capacitance.[1] A steel diaphragm exposed to cylinder pressure is mounted in one end of the pickup with the side away from the combustion chamber near one plate of an electrical condenser in such a way that the diaphragm itself forms the other plate. The design is such that the capacitance of this condenser is directly proportional to cylinder pressure. A high-frequency voltage applied to the condenser from an oscillator is then modulated by movement of the diaphragm to give a signal that can be amplified and fed to an oscilloscope. The capacitance type of pickup has the advantage that it can be used for either static or dynamic pressures, and the pressure calibration can thus be checked easily and accurately. The carrier wave from the oscillator also provides a convenient time scale for correlating pressure with crank angle. It cannot be used for investigating the high-frequency pressure waves of detonation, however.

Appearance of Indicator Diagrams. It was stated in Chap. II that the later types of diesel engine as well as all types of spark-ignition engine operate on a thermodynamic cycle closely approximating the constant-volume cycle. Just how closely the actual cycles for these engines approach such an ideal can be shown most clearly by plotting representative indicator diagrams on logarithmic coordinates. With this form of representation, the compression and expansion processes plot as straight lines having slopes equal to their respective polytropic exponents. Thus all four sides of the ideal diagram plot as straight lines, and deviations of the actual diagram from the ideal can be seen very easily. Figure 67 shows a representative spark-ignition engine indicator diagram plotted in this fashion. It is superimposed on the diagram for the corresponding ideal constant-volume cycle. Figure 68 shows an indicator diagram for a representative diesel engine, again plotted on logarithmic coordinates. Note how closely the actual cycles approach the form of the ideal constant-volume cycle in both cases, especially in so far as the combustion and exhaust processes are concerned.

Use of Logarithmic Indicator Diagrams. While the area of an indicator diagram plotted on logarithmic paper cannot be used as a measure of the net work accomplished during the cycle, the logarithmic plot provides a powerful technique for analysis work.[2] It has many uses in addition to those evident in the above discussion. One nice application of the technique is evident in Figs. 67 and 68. The experimental error in determining the individual points for a diagram may be such that it would be difficult to draw a mean curve through

them if they are plotted on linear coordinates. The same points plotted on logarithmic coordinates define a straight line much more accurately.

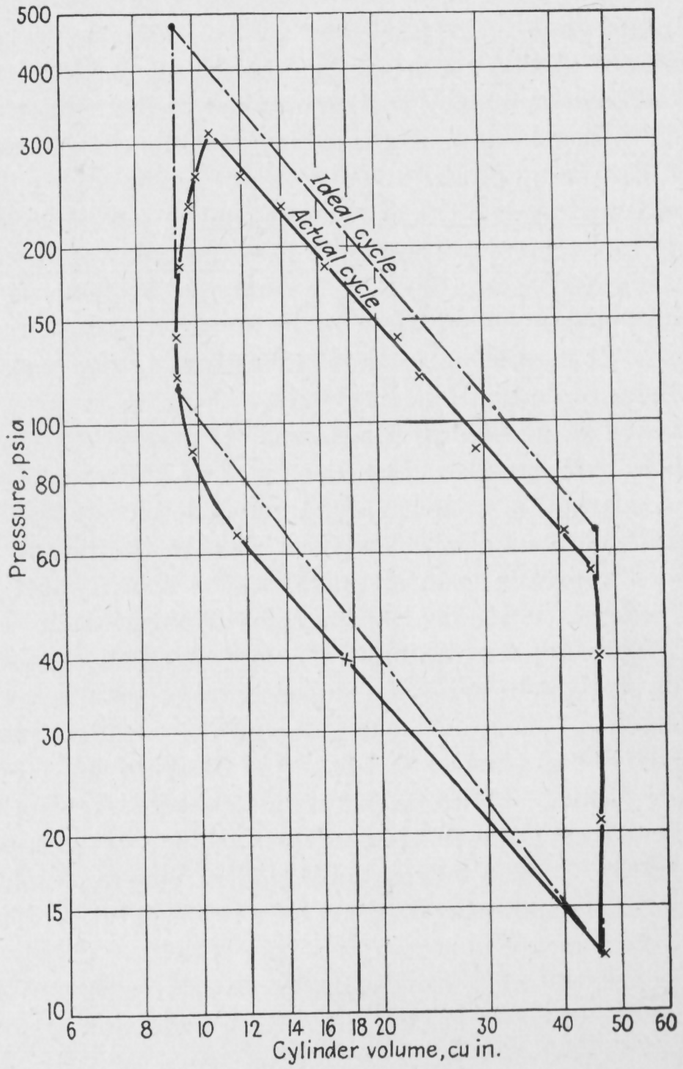


FIG. 67. Indicator diagram for a CFR fuel test engine superimposed on the diagram for the corresponding constant-volume cycle. (*Case Institute of Technology.*)

Survey of Differences between Ideal and Actual Cycles. In examining a set of diagrams such as those in Fig. 67, a number of sources of difference between the ideal and actual cycles becomes evident. In determining their nature and effects, two assumptions may be made that hold quite closely in practice. The first is that, as demonstrated in reference 8, Chap. IV, combustion of any given portion of

the charge may be considered as complete very shortly after passage of the flame front. Thus virtually all the heat energy ideally available from combustion is released a relatively few degrees of crank angle after the flame front has traversed the cylinder. The second is that the temperature, pressure, and volume of the gases in the combustion

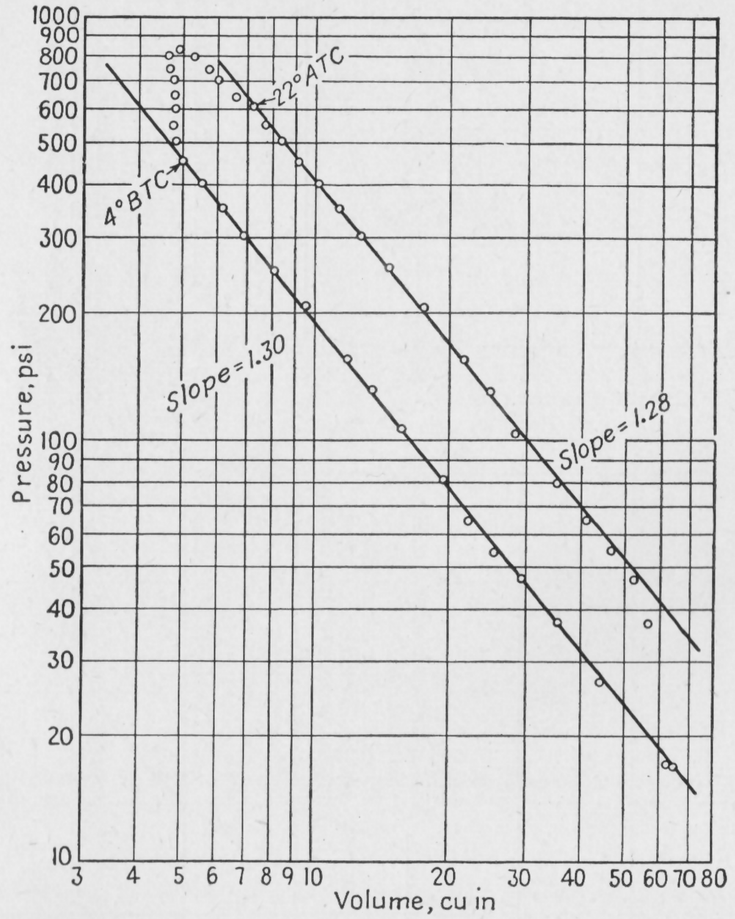


FIG. 68. Indicator diagram for a truck diesel engine plotted on logarithmic coordinates. [*Withrow (discussion), Trans. SAE, p. 773, Vol. 53, 1945.*]

chamber can be related to each other and the heat of combustion by the basic thermodynamic relationships if allowances are made for the effects of variable specific heats and dissociation.[3,4] *Therefore the only factors that can cause the actual process to deviate from the corresponding ideal are the following:*

1. "Rounding" of the corners on the left side of the indicator diagram by combustion of much of the charge either before or after top center

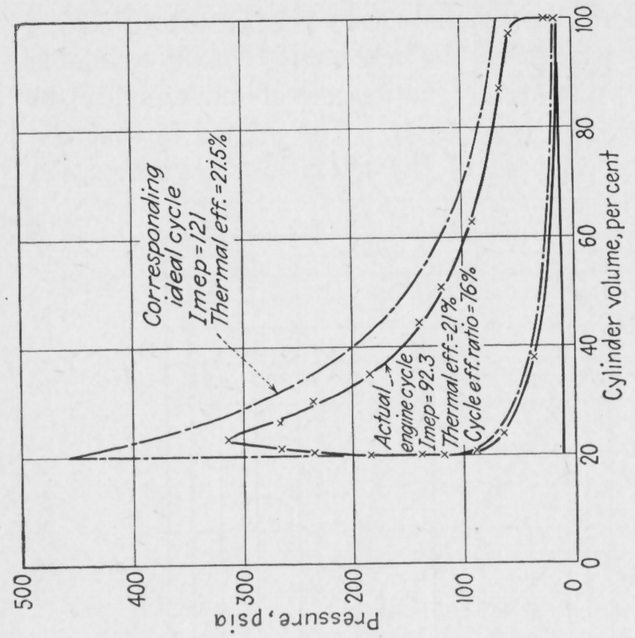
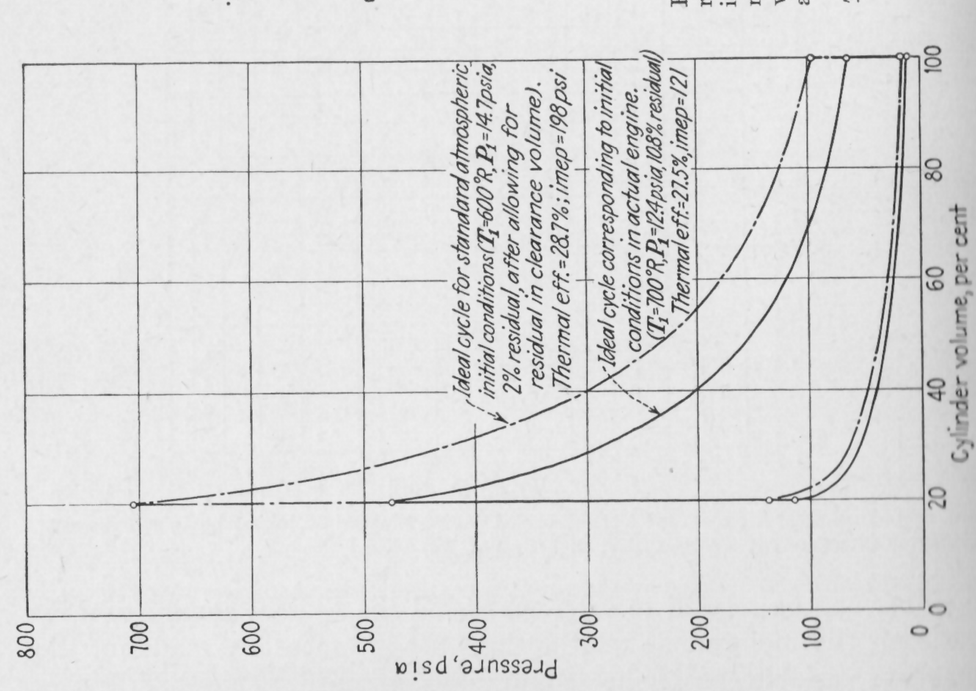


FIG. 69. Comparative *P-V* diagrams for three closely related cycles, *i.e.*, an idealized cycle with atmospheric initial conditions before the fresh charge is mixed with the residual exhaust gas; the idealized constant-volume cycle with the same initial conditions as an actual cycle; and an actual cycle. CFR fuel test engine, compression ratio = 5.0:1, fuel-air ratio = 0.0814. (*Case Institute of Technology.*)



2. Rounding of the corners on the right side as a result of the considerable period of time required for the exhaust-blowdown process.

3. The diluting and heating effect of the exhaust that remains in the cylinder and mixes with the fresh charge for the succeeding cycle

4. Heat transfer to or from the combustion-chamber walls

5. Air pumping losses

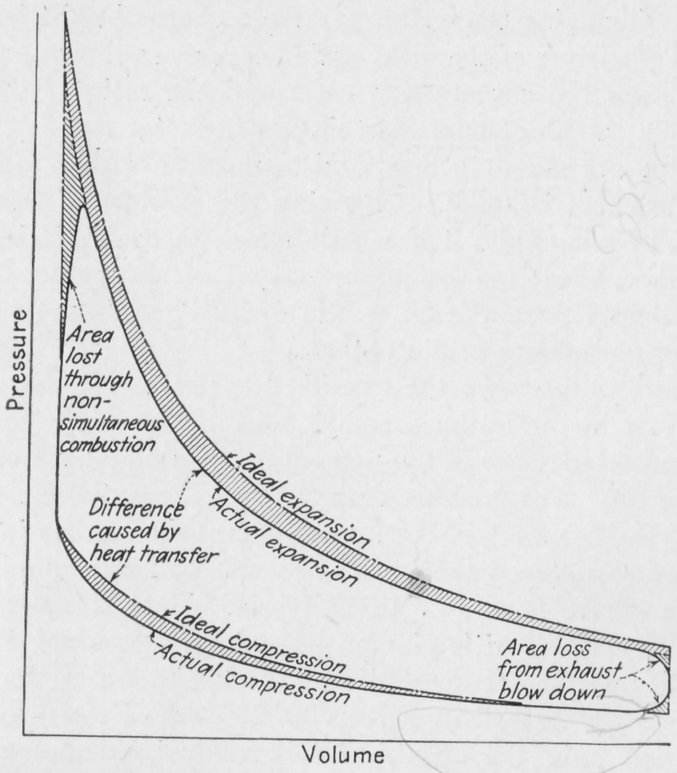


FIG. 70. Nature of differences between the indicator diagrams for ideal and actual constant-volume cycles.

The nature and magnitude of the various types of loss can be seen more readily by examining the data of Fig. 67 as it is replotted on linear coordinates in Fig. 69 and comparing it with a cycle computed in the manner indicated at the end of Chap. III. Two sets of diagrams are given—one set comparing the actual cycle with its idealized counterpart having the same weight of charge per cycle and the same initial pressure, and a second set comparing the latter with a cycle having the conditions at the beginning of the compression stroke equal to those in the intake manifold except for allowances made for residual exhaust gas in the clearance volume. This procedure divides the losses enumerated in the preceding paragraph into two parts, viz., those losses affecting the *weight* of fresh charge induced each cycle,

and those affecting the thermodynamic *processes* that the working fluid undergoes. The former are determined by the volumetric efficiency of the engine (*i.e.*, its performance as an air pump), while the latter are determined by the degree to which the compression and expansion processes approach the ideal isentropics and how closely the combustion and exhaust events approach the ideal constant-volume conditions. Changing the *quantity* of fresh charge has little effect on the thermal efficiency of the ideal cycle because, as shown on page 35, this depends on the compression ratio and the ratio of the specific heats. While the amount of dissociation (a factor not considered in the derivation on page 35) does tend to increase with a reduction in pressure, the effect is small. Changing the *quantity* of fresh charge induced each cycle, while it has little effect on cycle efficiency, does have a marked effect on the power output of an engine, for engine output is directly proportional to the quantity of fresh charge supplied if other factors are held constant.

The difference between the two ideal cycles in Fig. 69*a* is caused by a reduction in the initial pressure, an increase in the initial temperature, and an increase in the percentage of residual exhaust gas for the actual cycle. The pressure drop through the test-stand air metering system, the carburetor, the intake manifold, and the intake port in this instance reduced the initial pressure at the beginning of the compression stroke from 14.7 to 12.4 psia. Heat transfer from the cylinder walls coupled with a larger percentage of residual exhaust gas combined to raise the temperature at the beginning of the compression stroke to 700°R instead of the 600°R obtained when the calculations included only the effects of the residual exhaust gas in the clearance volume.

The causes for the differences between the two diagrams in Fig. 69*b* are indicated in Fig. 70. They result from differences between the assumed ideal thermodynamic processes and the actual processes that take place in the engine.

A similar comparison between the actual cycle in a diesel engine and its idealized constant-volume counterpart discloses that the factors causing differences between the two are the same as in a spark-ignition engine except that an additional loss is introduced by the fact that in the diesel combustion may not be complete at some definite point shortly after top center. While most of the fuel charge burns in even less time than would be the case in a corresponding spark-ignition engine, under heavy load conditions a portion of the fuel may continue to burn during the expansion stroke as well. Careful design of the combustion chamber and fuel injection system can greatly reduce this loss.

LOSSES ASSOCIATED WITH COMBUSTION RATE

Effect of Combustion Rate. Figures 71 and 72 show pressure–crank-angle and pressure-volume diagrams for four different combustion rates. One is for instantaneous combustion at top center as assumed in the idealized constant-volume cycle, a second is for a high, a third for a moderate, and a fourth for a low rate of combustion. While spreading the combustion process out over a considerable period of time entails a small power loss, as is evident in Fig. 72, it provides a very effective means of reducing the peak pressure. This is desirable in that it reduces both the stresses in engine parts and the rate of crankshaft acceleration and hence the severity of crankshaft vibration. The loss in indicator-diagram area from noninstantaneous combustion is only about 2 per cent under favorable conditions.

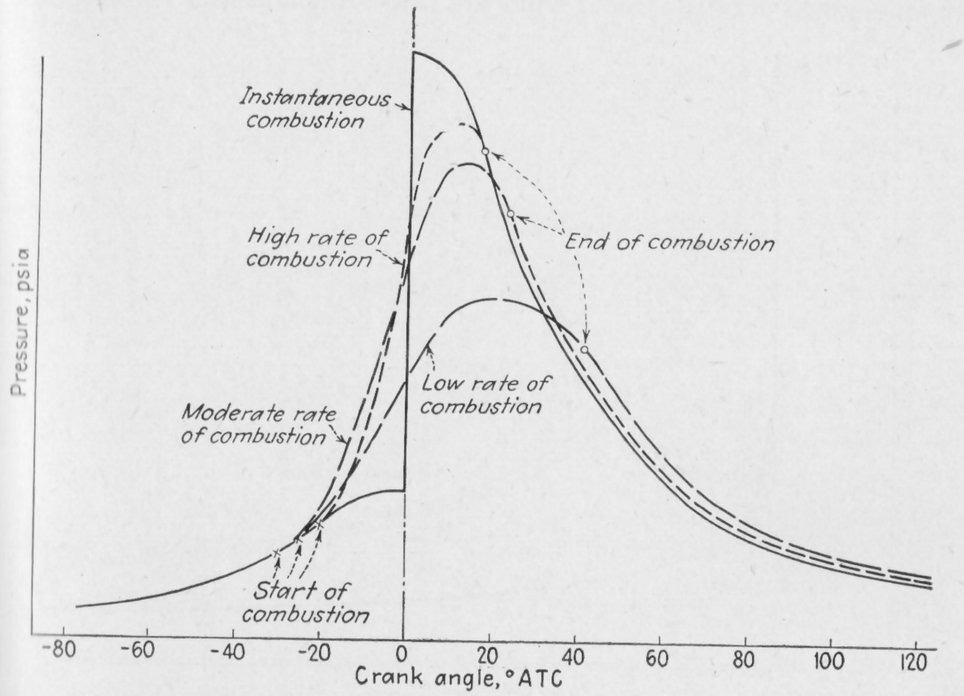


FIG. 71. Effect of combustion rate on the pressure–crank-angle diagram.

Factors Determining Combustion Rate. In a spark-ignition engine, a variety of factors influence the rate at which the charge is burned through their effects on the area of the flame front, the density of the charge in the flame front, or the flame speed. As pointed out

in Chap. IV, the flame speed depends on many operating variables and may vary considerably—especially with random variations in the amount and character of turbulence during combustion. Since this turbulence is the remainder of that induced during the intake stroke and depends on whether the eddies formed in the induction process tend to reinforce or extinguish one another, it is prone to change considerably from one cycle to the next. Figure 73 shows this tendency in an engine operating under what are ostensibly stabilized conditions.

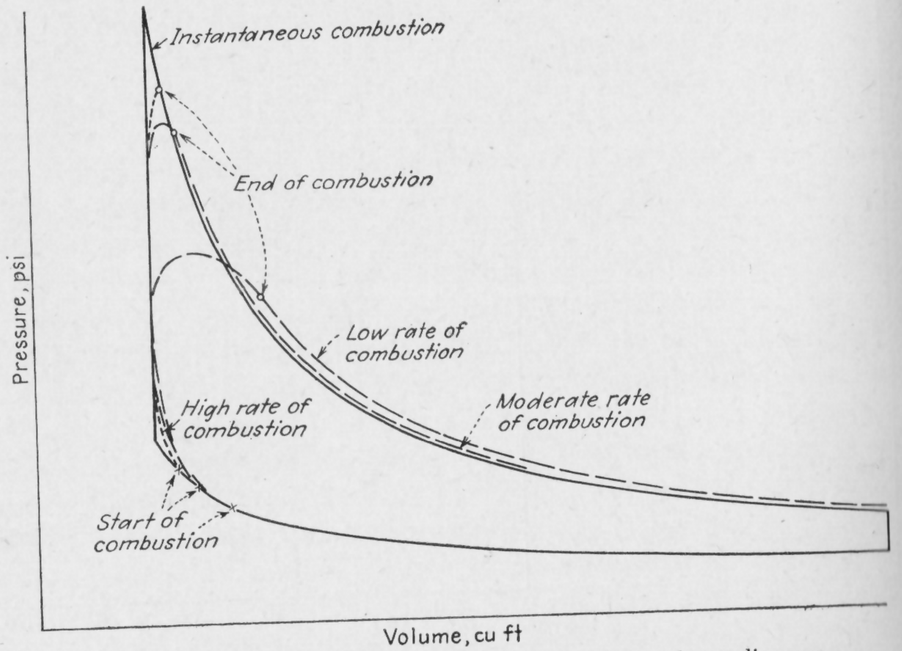


Fig. 72. Effect of combustion rate on the pressure-volume diagram.

While the variations in the pressure–crank-angle diagram appear large, reference to Figs. 71 and 72 discloses that very little cycle-to-cycle variation in work output will result from them in spite of the considerable variations in peak pressure. It is important for one to realize that these cycle-to-cycle irregularities occur, for they help to explain why something like detonation may happen only once in every 20 to 50 cycles in a given cylinder. Other phenomena such as preignition, misfiring, and backfiring are similarly affected.

Effect of Combustion-chamber Shape. The combustion rate for the cylinder charge as a whole at any given instant is directly proportional to the flame-front area. This, in turn, is almost purely a function of combustion-chamber geometry. The effect can be visualized readily by considering the three hypothetical combustion chambers in

Fig. 74. The first is a cone with the spark plug placed at the apex. As the flame front moves down the cone, its area increases as the square of the distance from the spark plug. If, on the other hand, the spark plug is placed in the base of the cone as in Fig. 74*b*, the flame-front area increases very rapidly until it touches the walls of the cone and then falls off as it moves toward the apex. The cylindrical combustion chamber of Fig. 74*c* gives still a different combus-

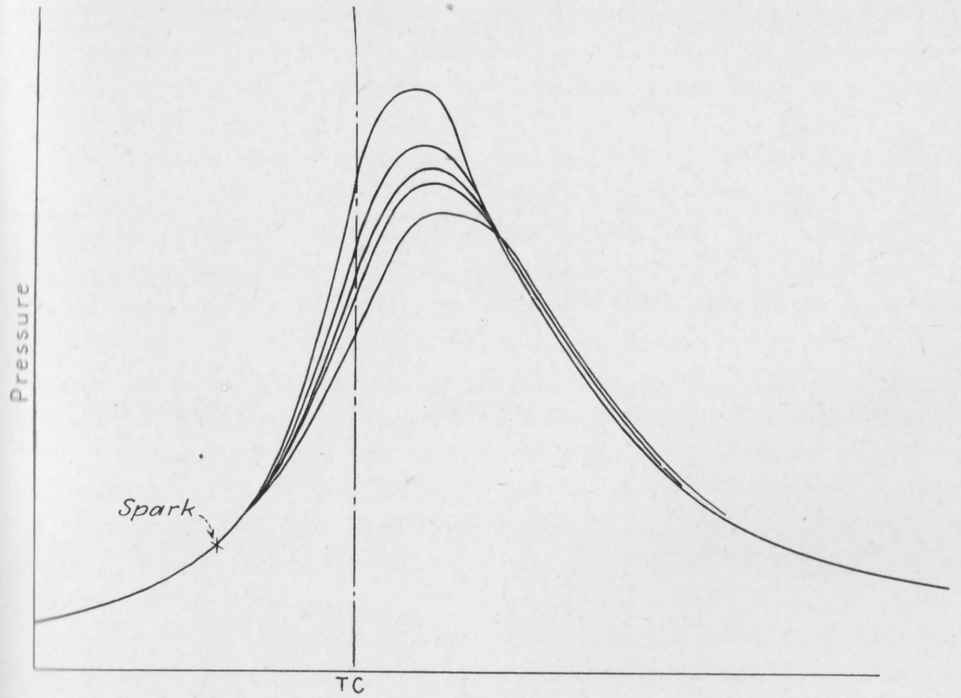


Fig. 73. Effect of cycle-to-cycle variations in combustion rate on the pressure–crank-angle diagrams for five consecutive cycles.

tion rate, for the flame-front area remains substantially constant after it has progressed out to the walls.

While the power-output and fuel-economy characteristics of a spark-ignition engine are not greatly affected by changes in combustion rate unless the combustion process is thereby spread out over too long a period, engine vibration and engine "smoothness" are very much dependent on the form of the combustion-rate curve. To obtain smooth engine operation, the third derivative of the pressure–crank-angle diagram should give a smooth and continuous curve.[5,6] Since the flame front must start from virtually a point source, the rate of increase in flame-front area is low and smooth for practically all combustion-chamber forms. Tapering off the latter portion of the

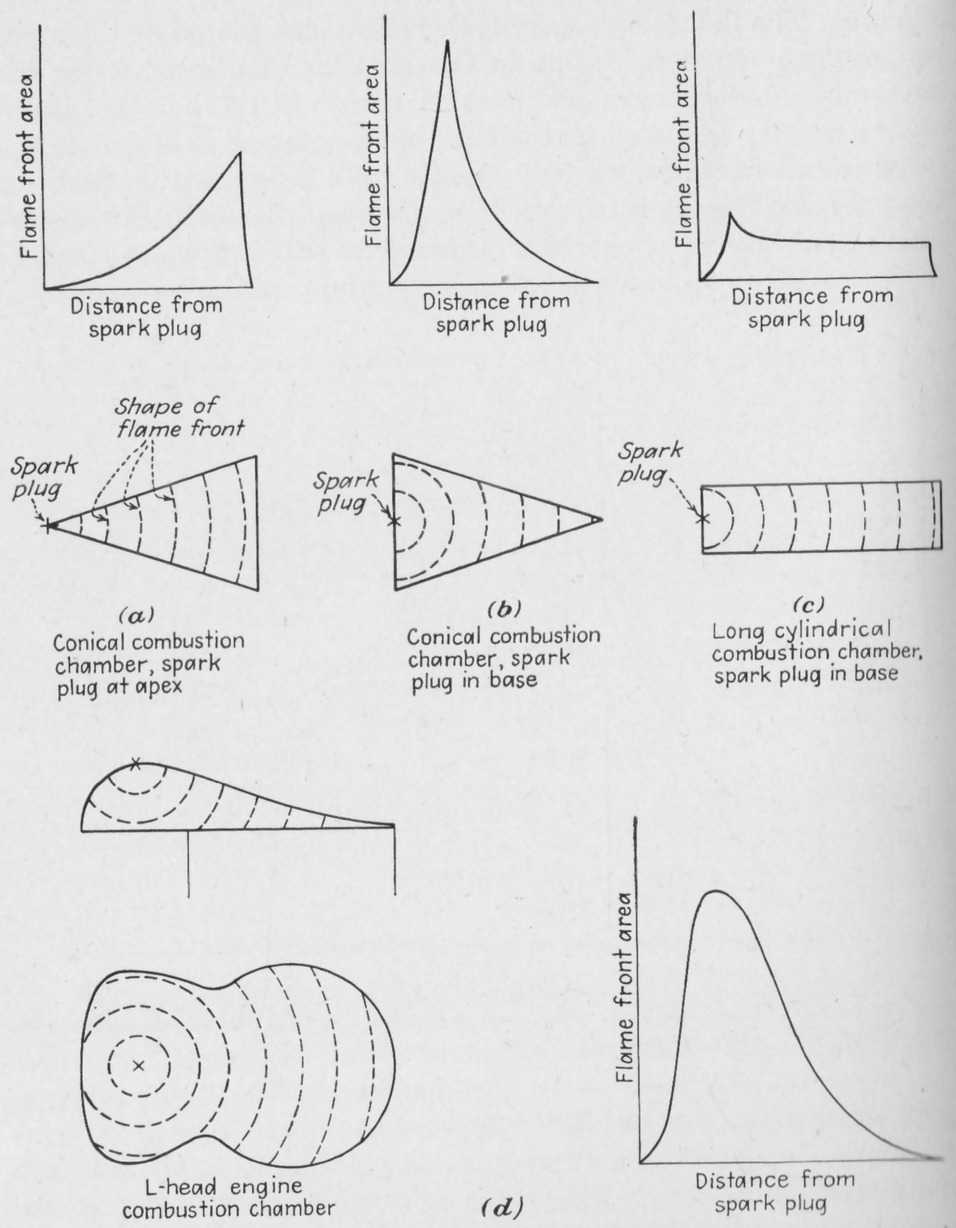


FIG. 74. Sections through three hypothetical combustion chambers and one commonly used form showing the effect of combustion-chamber shape and spark-plug location on the shape and area of the flame front at successive distances from the spark plug.

combustion process so that there will be a smooth rate of decrease and hence no abrupt change in the rate of crankshaft acceleration is sometimes difficult, however. The nature of the problem is indicated by the curves for flame-front area vs. flame travel in Fig. 74. The conical combustion chamber with the spark plug at the base of the cone would give a desirable form to the latter portion of the combustion-rate curve because it would give a gradual reduction in the rate of energy release. On the other hand, if the spark plug were placed at the apex of the same conical combustion chamber, this would give a very undesirable curve, for the combustion rate would increase rapidly throughout the combustion process and then drop abruptly to zero. The effect is even greater than is indicated by the flame-front-area curve, for compression of the unburned charge during combustion increases its density so that the density of the burning gas is about three times as great during the latter phases of combustion as during the initial phases.

The best form of combustion chamber for spark-ignition engines has been found to be one in which the flame is permitted to spread out in hemispherical form during the early stages of combustion to get the process under way as quickly as possible. The chamber should be so contoured that during the middle and latter stages of combustion the flame-front area will be progressively reduced until it is quite small during the last stages of combustion. The combustion chamber of Fig. 74*d* with the dotted lines indicating successive flame-front positions shows how these principles are applied in a typical automotive engine. Note how smooth the curve of "flame-front area vs. distance from the spark plug" is for this combustion-chamber shape. A smooth pressure–crank-angle diagram and hence a smooth-running engine will result.

Combustion Rate in a Diesel. While the initiation and propagation of combustion in a diesel differ considerably from the corresponding processes in a spark-ignition engine, the rate of combustion is also greatly dependent on air turbulence. Because of this, pressure–crank-angle diagrams for diesel cylinders are also inclined to show cycle-to-cycle variations in combustion rate, although much less so than in spark-ignition engines. In general, the combustion rate in a high-speed diesel is much higher than in a spark-ignition engine.[7] This high combustion rate coupled with the higher peak pressures brought about by the higher compression ratios of the diesel make heavier parts necessary and cause the operation of a diesel to be "rougher"—*i.e.*, characterized by more intense high-frequency torsional vibrations—than for a similar spark-ignition engine.

LOSSES ASSOCIATED WITH VALVING OF THE WORKING FLUID

Valve Timing. The very serious acceleration problems associated with the opening and closing of intake and exhaust valves are such

that a considerable period of time must be provided for these events. It has been found that, for optimum engine performance, the valves must open earlier than they would in the ideal cycle and close later. Figure 75 shows valve opening area plotted against crank angle for both the ideal and an actual cycle. The difference between the two curves tends to give some loss in output, which depends in part on the valve timing.

The valve timing for a four-cycle engine is generally expressed in the form shown in the table in Fig. 75. The terms *before top center*,

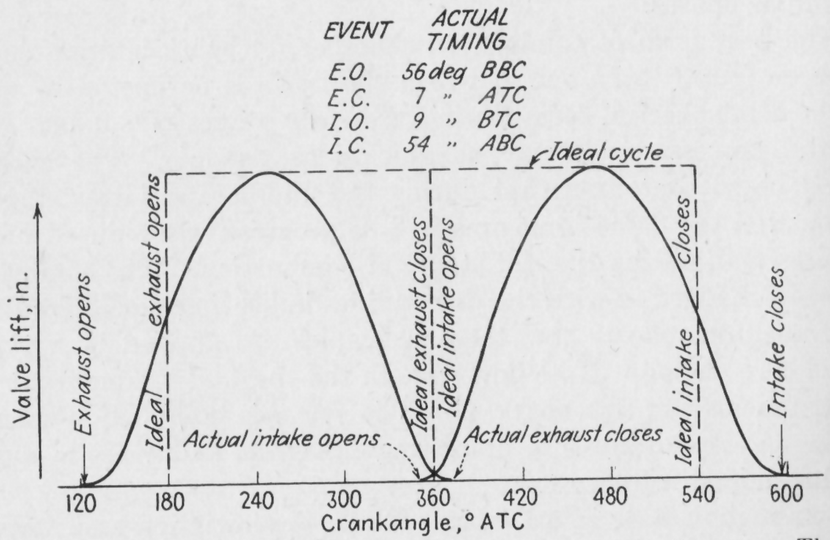


FIG. 75. Comparison of ideal and actual valve-lift crank-angle diagrams. The diagram for the ideal cycle is shown in dotted form. The solid curves show the actual diagram for a typical passenger-car engine.

after top center, etc., are generally abbreviated as in Fig. 75 to BTC, ATC, etc. Note that both valves are open during the interval between the intake-valve opening and the exhaust-valve closing. This period is referred to as valve *overlap*. It is usually small for low-speed engines but relatively large for high-speed engines.

Effect of Exhaust Blowdown. Relieving the pressure in the cylinder by exhausting the working fluid requires an appreciable period of time. Even if it were possible for the exhaust valve to be opened instantaneously, an appreciable period would be required for the gas in the cylinder to escape and relieve the pressure. Since the exhaust valve must be opened at a finite rate, the exhaust-blowdown process requires an even greater period of time and hence results in a rounding of the corners at the right end of the indicator card.

Pumping Losses. To be significant, the "pumping-loop," or low-pressure, portion of the indicator card must be plotted against a much larger pressure scale, as in Fig. 76. The term *pumping loop* is used because the area of the loop is proportional to the net work done by the piston in drawing the gas into the cylinder and expelling it. Unless the engine is supercharged, the pressure in the cylinder during the intake stroke will be below atmospheric. During the exhaust stroke, on the other hand, it will usually be above atmospheric. Thus the piston must do work during both strokes, and the work represented by the area of the pumping loop should be deducted from that of the

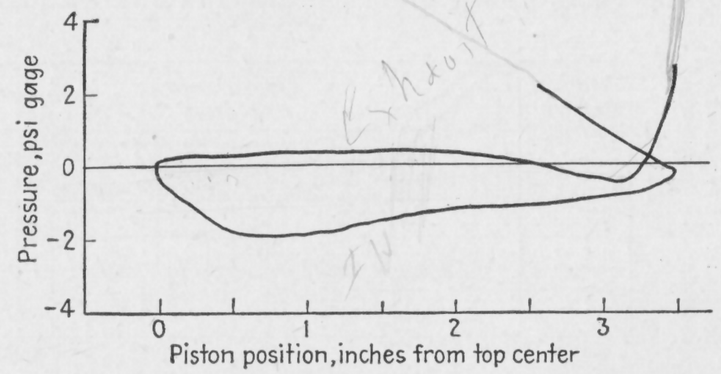


FIG. 76. Pumping loop of a representative four-cycle engine.

power loop if the net work derived from the cycle as a whole is to be obtained. In general, the area of the pumping loop is small, seldom exceeding 1 per cent of that of the high-pressure loop under full-load conditions. Under part-throttle conditions in spark-ignition engines, of course, it is proportionally much larger.

HEAT LOSSES TO COMBUSTION-CHAMBER WALLS

Heat Transfer as a Function of Crank Angle. The energy released by combustion first goes into heating the gases in the cylinder, and then some is utilized in doing work on the piston during the expansion stroke. During combustion, the expansion stroke, and the exhaust process, some of this heat flows to the combustion-chamber walls. The instantaneous distribution of the chemical energy of combustion between the chemical energy in the fuel, the heat energy of the gas, heat losses to the combustion-chamber walls, and useful work done on the piston is shown in Fig. 77 as a function of crank angle.

In examining Fig. 77 it should be noted that heat transfer during the exhaust process occurs after the power stroke and hence has no effect on the useful work done on the piston. Similarly, heat transfer

during the latter portion of the power stroke has no effect on the useful work accomplished during the first portion of the expansion. Thus, from the standpoint of the indicator diagram, the important phase of the heat-transfer process is during combustion and the early part of the expansion stroke. This loss is difficult to determine experimentally because it cannot be taken directly from the total heat flow to the coolant of an engine but must be estimated from a comparison of calculated and actual indicator diagrams.

Effect of Surface-volume Ratio on Heat Loss. One of the most important factors affecting the magnitude of the heat loss near top center is cylinder size. Since the heat-transfer rate varies directly with

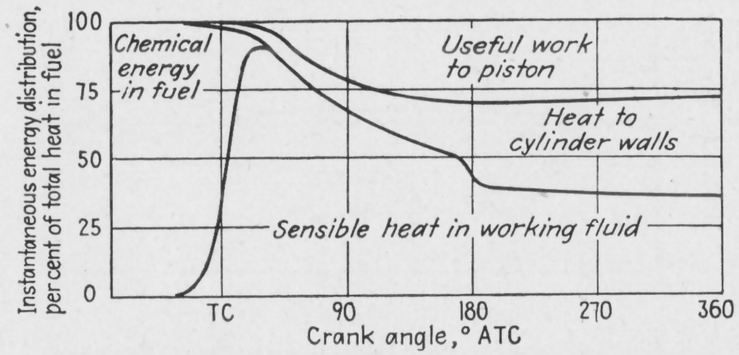


FIG. 77. Instantaneous distribution of the energy of combustion between the chemical energy in the fuel, the sensible heat in the working fluid, useful work delivered to the piston, and heat loss to the combustion chamber and exhaust-port walls.

the area of the exposed surface, the greater the amount of combustion-chamber surface per unit weight of charge, the greater the percentage of heat in the charge that will be lost to the walls. The surface-volume ratio may be derived readily since, for any particular shape of chamber,

$$\frac{\text{Surface}}{\text{Volume}} = \frac{bd^2}{d^3} = \frac{b}{d}$$

where d is the cylinder diameter and b is a constant that depends on the form of the combustion chamber. Thus it is evident that the percentage of the total heat that is lost from the charge is inversely proportional to cylinder size for a series of geometrically similar cylinders. The relation must be modified, of course, if the combustion-chamber shape is altered or if the ratio of the bore to the stroke is changed.

As was pointed out above, the loss in indicator-diagram area caused by heat transfer to the combustion-chamber walls not only is difficult to determine but depends on many operating variables. Its

magnitude at full load and normal engine speeds may cause a loss of as little as 3 per cent in indicator-diagram area in a 6.125-in.-bore air-cooled cylinder with a hemispherical combustion chamber. On the other hand, depending on the speed and load, the loss in diagram area from heat transfer commonly runs from 10 to 15 per cent in L-head automotive engines, largely because of their higher surface-volume ratios.

The surface-volume ratio of the combustion chamber may also influence the thermal efficiency of an engine through its effect on combustion. As was pointed out in Chap. IV, the ignition lag and hence

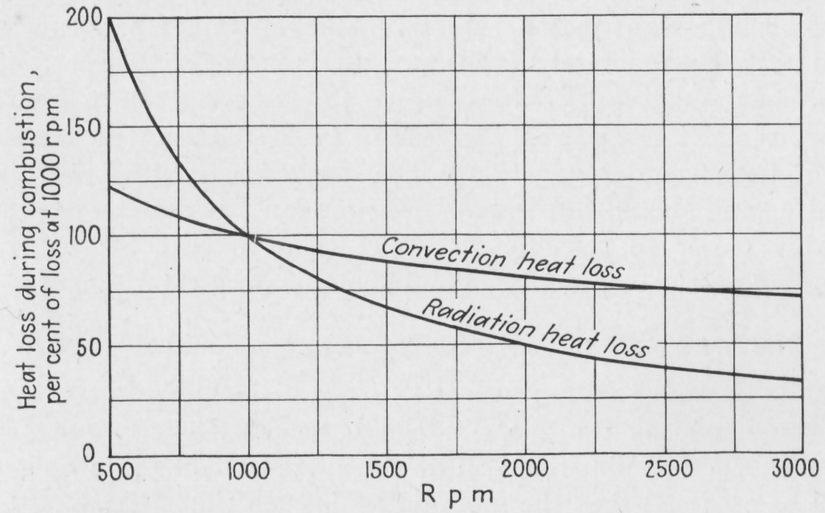


FIG. 78. Effect of rpm on the two types of heat loss prominent during combustion. Values are given for each relative to its respective loss at 1,000 rpm.

the combustion rate in a diesel are very much dependent on the temperature of the air in the cylinder during injection. The amount of heat lost to the combustion-chamber walls during the latter portion of the compression process increases so rapidly as cylinder size is reduced that proper combustion at light loads and low speeds becomes a problem in a diesel engine if the cylinder bore is reduced below about 6 in.

Effects of Metal Temperature on Heat Loss. Fundamental heat-transfer considerations indicate that heat flows from the charge to the combustion-chamber walls both by radiation and by convection. The radiant-heat flow rate should vary as the fourth power of the temperature difference between the gas and the combustion-chamber walls, while the convection-heat flow rate should vary as the first power of that temperature difference. In either case, increasing the cylinder-

metal temperature tends to reduce the temperature differential and hence the heat loss and therefore increases the useful output of the cycle.

Effects of RPM on Heat Loss. The most important portion of the heat loss from the standpoint of engine output is that occurring during combustion. While the proportion of the heat transferred by radiation is quite small during most of the cycle, radiation appears to play the major role as the heat-transfer mechanism during combustion. Since the amount of heat transferred through radiation is directly proportional to time, the loss in power occasioned by radiation should be inversely proportional to the rpm. The heat lost through convection is also an important factor. As will be shown in Chap. XI, Cooling, it will vary approximately as the two-thirds power of the speed at any given mean effective pressure (mep). Curves are given in Fig. 78 to show the effect of rpm on the loss in cycle efficiency as caused by both radiation and convection. The curve for the total loss for an actual engine should fall somewhere between the two curves shown, probably closer to that for radiation than to that for convection depending on the engine design and the operating conditions.

FACTORS AFFECTING CYCLE OUTPUT AND EFFICIENCY

While there are many items that might be included in this section, only those relating the actual cycle to the ideal cycle under conditions of normal combustion will be considered. Thus in all instances it will be assumed that the fuel quality is such that no detonation or preignition occurs and that the engine is in good mechanical condition and hence free of losses caused by leaky valves, worn piston rings, etc. All the results are presented in terms of indicated output so that losses from mechanical friction need not be considered. Inasmuch as the fhp is comparatively large at light loads, curves showing the effects of operating variables on bhp and bsfc may look somewhat different from corresponding curves for ihp and isfc. The relations between operating variables and brake and indicated outputs will be covered in the chapter on engine performance.

Effect of Compression Ratio. Since it was demonstrated in Chap. II that the efficiency of the ideal constant-volume cycle depends primarily on compression ratio, it is interesting to compare the effects of this factor on the actual cycle with its effects on the ideal cycle. Figure 79 shows curves for the thermal efficiency, imep, peak pressure, and isfc for both the ideal constant-volume cycle and an actual engine. The engine data were obtained with the same basic model of engine

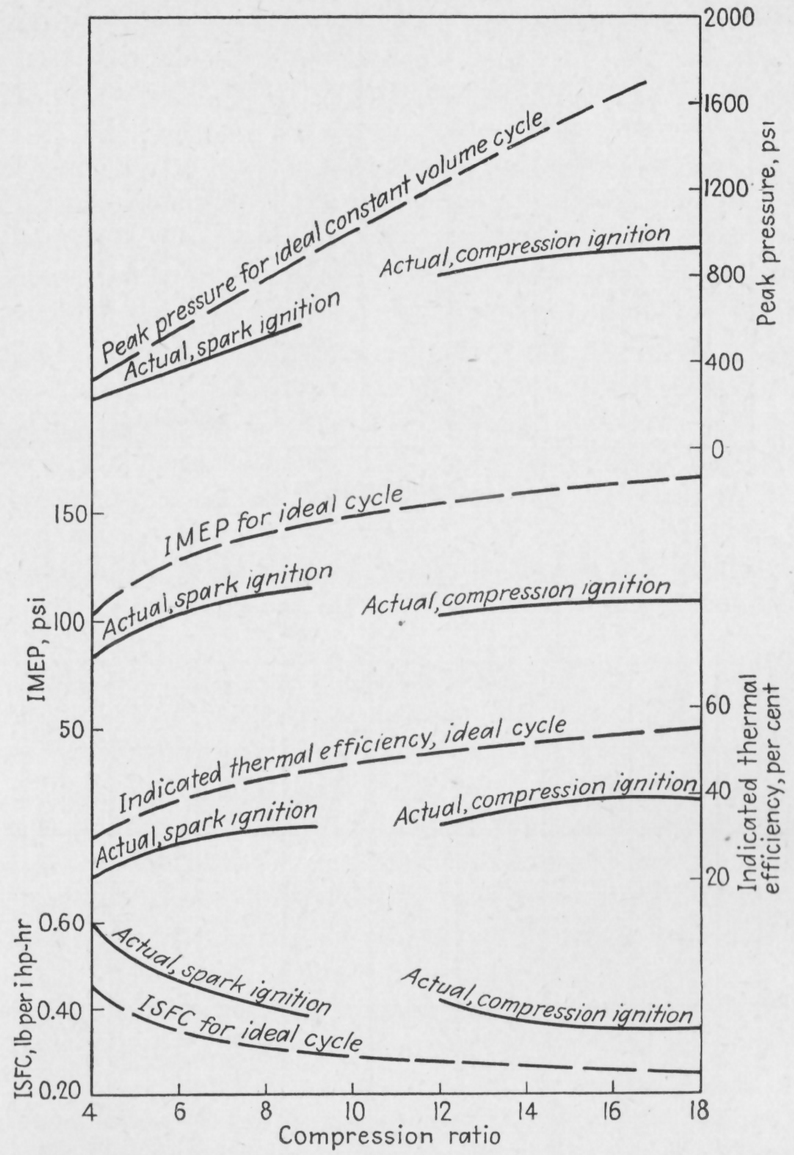
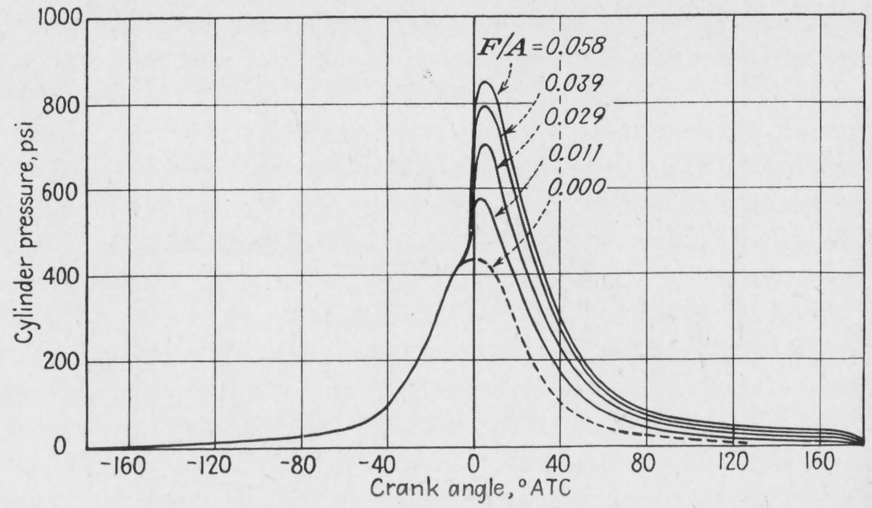


FIG. 79. Effect of compression ratio on both the actual cycle in a CFR single-cylinder fuel test engine, and the ideal constant-volume cycle computed for the same weight of charge per cycle as was present in the actual engine. Engine bore = 3.25 in.; stroke = 4.5 in.; fuel-air ratio = 0.0605; and rpm = 1,200. The amount of residual exhaust was estimated to be 10 per cent. (*Data from reference* 10 *and Case Institute of Technology.*)

fitted with two types of cylinder, one for spark-ignition operation at low compression ratios and one for compression-ignition operation at high compression ratios.[10] While other engine types might give curves of a somewhat different shape, those in Fig. 79 are reasonably representative of automotive-type engines. Note that the curves for



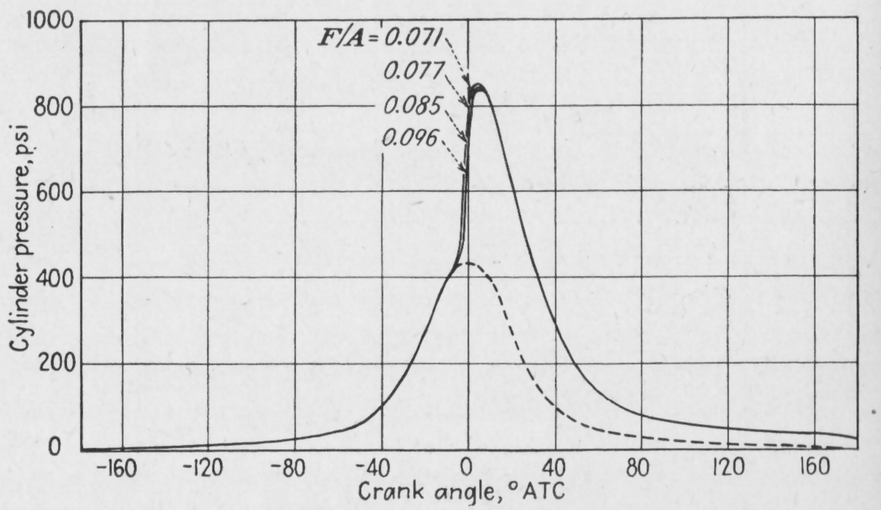


FIG. 80. Effect of fuel-air ratio on pressure–crank-angle diagrams for a diesel engine. (*Rothrock and Waldron, NACA Tech. Rept.* 545, 1936.)

the actual engine parallel those for the ideal cycle, the ratio of the former to the latter defining a cycle-efficiency ratio of about 75 per cent, a low value caused by the low rpm of this specialized type of fuel test engine.

The calculations for the ideal-cycle data were based on the actual temperature and pressure in the engine cylinder at the beginning of the compression stroke. This was done to separate the effects of

cycle-efficiency ratio from those of volumetric efficiency.[9] The latter will be discussed in a later section.

Effect of Fuel-Air Ratio. It was shown in Chap. II that the only factor affecting the efficiency of the ideal cycle other than the compression ratio is the ratio of the specific heats of the working fluid. In conventional engines using petroleum fuels, both the composition of the products of combustion and the temperature range for the cycle depend on the fuel-air ratio. This factor may cause a change in the ratio of the specific heats of 20 per cent or more. Since the combustibility characteristics of commercially available fuels limit the compression-ratio ranges of most interest to between 6.0:1 and 7.5:1 for spark-ignition engines and between 13.5:1 and 18:1 for diesel engines, the effects of fuel-air ratio can be shown quite adequately by data for just two compression ratios, one for each of the two engine types.

The effect of fuel-air ratio on pressure–crank-angle diagrams for a diesel engine is shown in Fig. 80[11], while the thermal efficiency of both the ideal constant-volume cycle and a representative diesel engine is shown in Fig. 81. Data are also given for the imep, isfc, and exhaust temperature. The actual engine-performance curves parallel those for the ideal cycle quite closely except at the very low fuel-air ratios, where so little fuel is injected that it does not burn properly. Heat losses near top center and piston motion during the combustion and exhaust-blowdown events account for the differences between the ideal and the actual curves. The effect of heat losses on the exhaust temperature is so great that the actual exhaust temperature is only about one-half the ideal value, a condition indicated by Fig. 77.

A similar set of curves for a spark-ignition engine is shown in Fig. 82. The difference between the curves for the ideal and the actual cycles is relatively small in the range in which the flame speed is high but becomes larger very rapidly as the mixture is leaned out beyond the chemically correct value, where the flame speed falls off severely. The loss in indicator-diagram area resulting from the low flame speed in lean mixtures may be deduced from the pressure–crank-angle diagrams of Fig. 71.

The fuel-air ratio is one of the principal factors subject to variation in spark-ignition engine operation, and hence the basic type of curve shown in Fig. 82 is one of the most important used in engine test and development work. It is known as a *mixture-loop* or a *mixture-control curve.* The bhp and bsfc are usually plotted instead of the imep and isfc, either fuel-air ratio or fuel flow being employed as the abscissa.

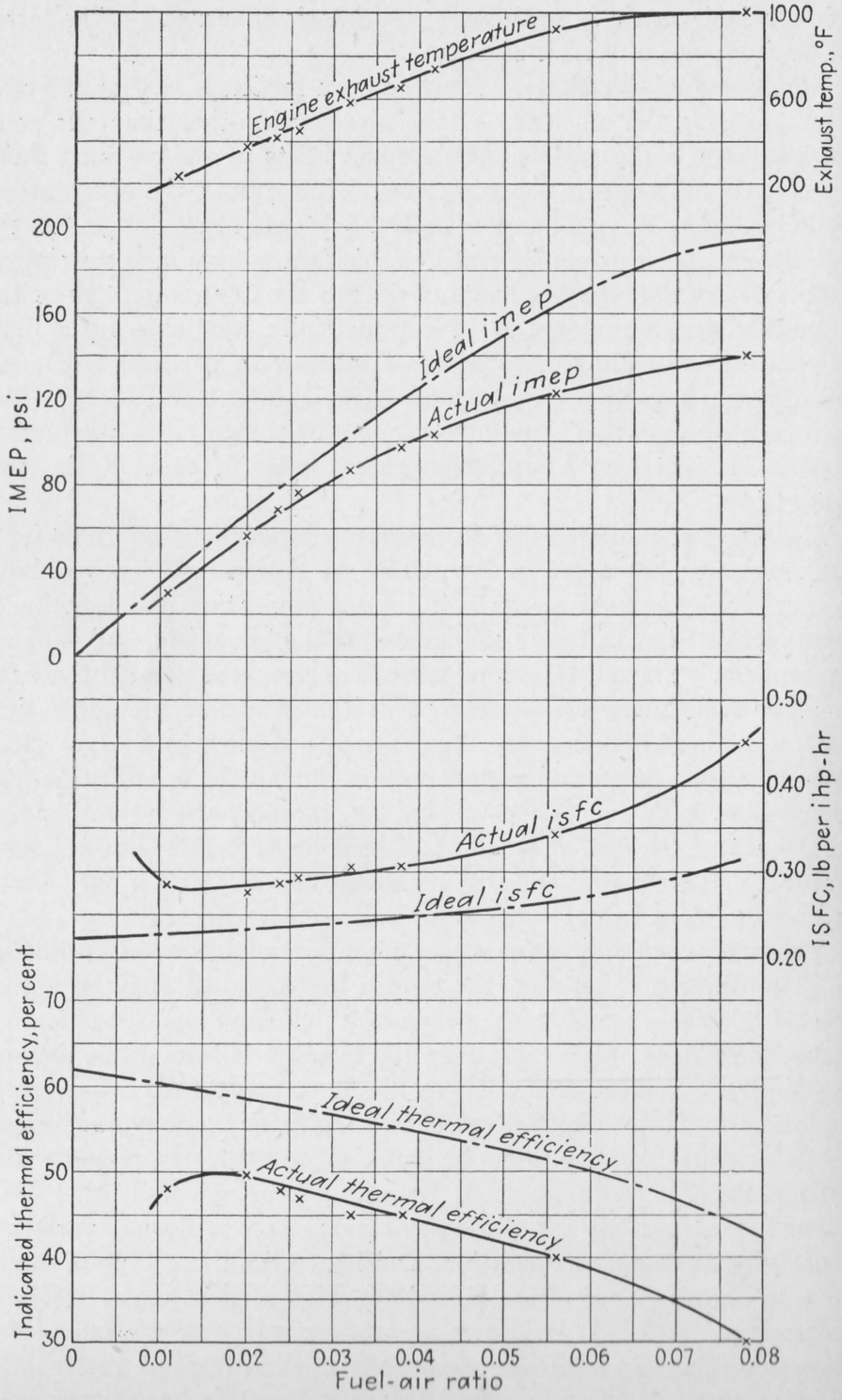


FIG. 81. Effect of fuel-air ratio on both the ideal constant-volume cycle and the corresponding actual cycle as determined for a Hercules four-cylinder DOOB truck diesel engine at 1,200 rpm. (*Case Institute of Technology.*)

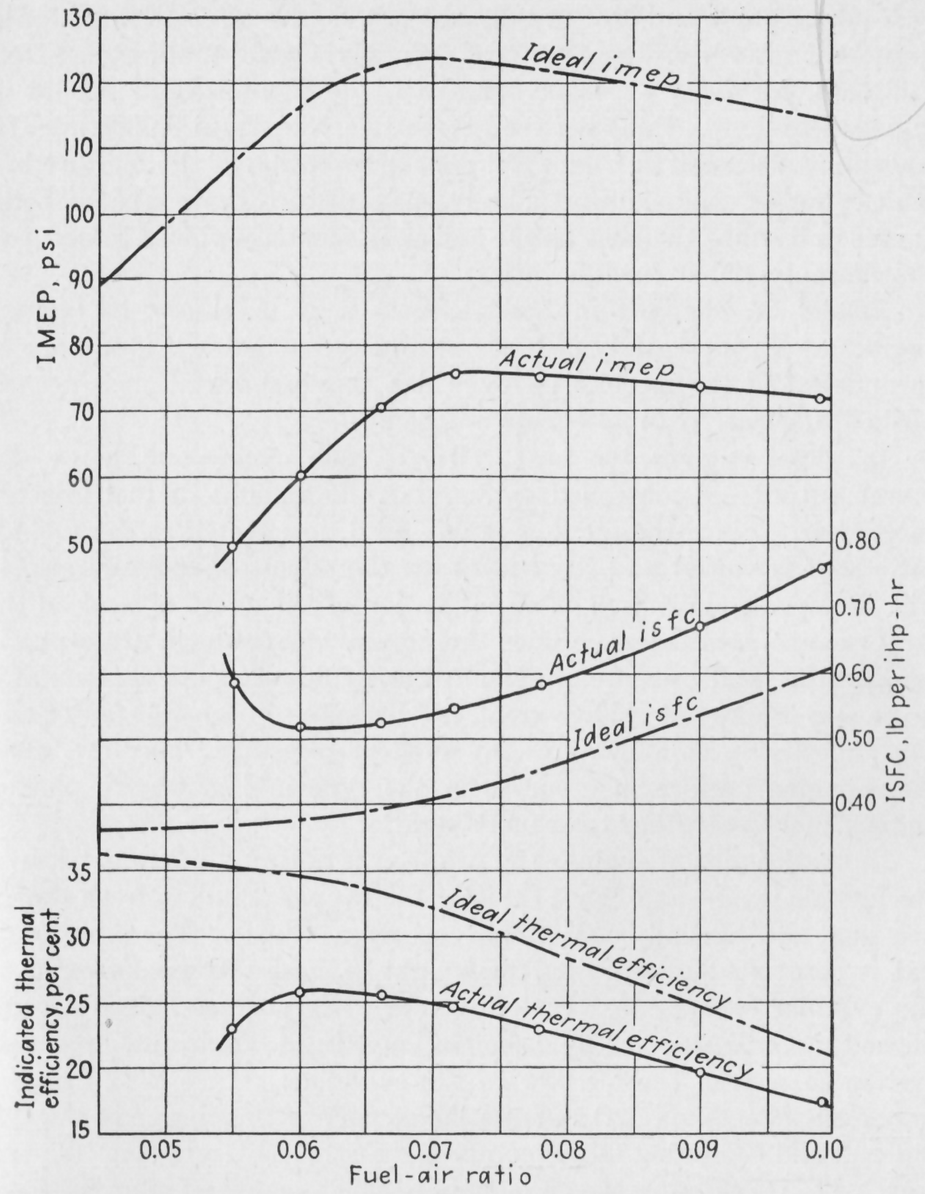


FIG. 82. Effect of fuel-air ratio on both the performance of a CFR single-cylinder engine and a constant-volume cycle having initial conditions equal to those in the CFR engine intake port. (*Case Institute of Technology.*)

Since the bhp and bsfc are adequate measures of power and fuel economy for most purposes, the thermal efficiency is seldom calculated. In using curves such as this the fuel-air ratio for maximum power is referred to as the *best-power* fuel-air ratio, while that for minimum sfc is referred to as the *best-economy* value. Note how much the fuel consumption increases with an increase in fuel-air ratio beyond the

best-power point and how rapidly the power falls off with a reduction in fuel-air ratio below that for best economy. One would expect from this that, for most types of operation, the fuel-air ratio should be adjusted to give a value between these two, viz., from 0.060 to 0.075. As will be discussed in Chap. IX, poor distribution of the mixture in a multicylinder engine unfortunately may distort the shape of these curves and cause the best power and best economy points to occur at considerably richer fuel-air ratios.

Effects of Changes in Load. Since it is necessary to operate engines at a wide variety of loads at any given speed, it is important to understand the mechanism by which the load variation is accomplished, together with its effects on the cycle.

In diesel engines the heat released each cycle—and hence the power output—is controlled by varying the amount of fuel injected each cycle. Since the air flow is left fixed, in effect it is the fuel-air ratio that is varied and the curves for the effects of fuel-air ratio in Fig. 81 serve equally well as an indication of the effect of load on the performance parameters. Since the normal variations in atmospheric temperature and pressure are small, the air flow to the cylinders at a given speed does not change greatly. Because of both this factor and the undesirable complications involved in measuring air flow, most diesel-engine performance curves of the type in Fig. 81 are plotted against fuel flow rather than fuel-air ratio.

In spark-ignition engines the problem is entirely different because the fuel-air ratio—and hence the heat release per pound of fresh charge—is kept substantially constant irrespective of load. The power output is controlled by varying the weight of fresh charge delivered to the cylinder each cycle. In four-stroke-cycle engines this is accomplished by *throttling, i.e.*, by closing a valve in the engine air induction system to restrict the air flow to the cylinders. Throttling has two important effects. (1) It reduces the density of the charge in the cylinder at the beginning of the compression stroke by reducing its pressure. (2) It increases the amount of exhaust-gas dilution of the fresh charge, partly because exhaust gas tends to blow back from the exhaust manifold through the cylinder into the intake manifold during the valve overlap period, and partly because the residual exhaust gas in the clearance volume expands to the lower pressure of the intake manifold. It is possible to visualize the blowback effect at low engine speeds if one realizes that the pressure in the intake manifold may be 15 in. Hg or more below atmospheric, while the pressure in the exhaust manifold usually runs somewhat above atmospheric. The residual

exhaust mixes with the fresh charge and reduces the amount induced both by dilution and by raising its temperature as shown in Tables 8 and 9 (pages 71 and 72) which are for full— or wide-open—throttle operation. The effects at part throttle can be far greater.

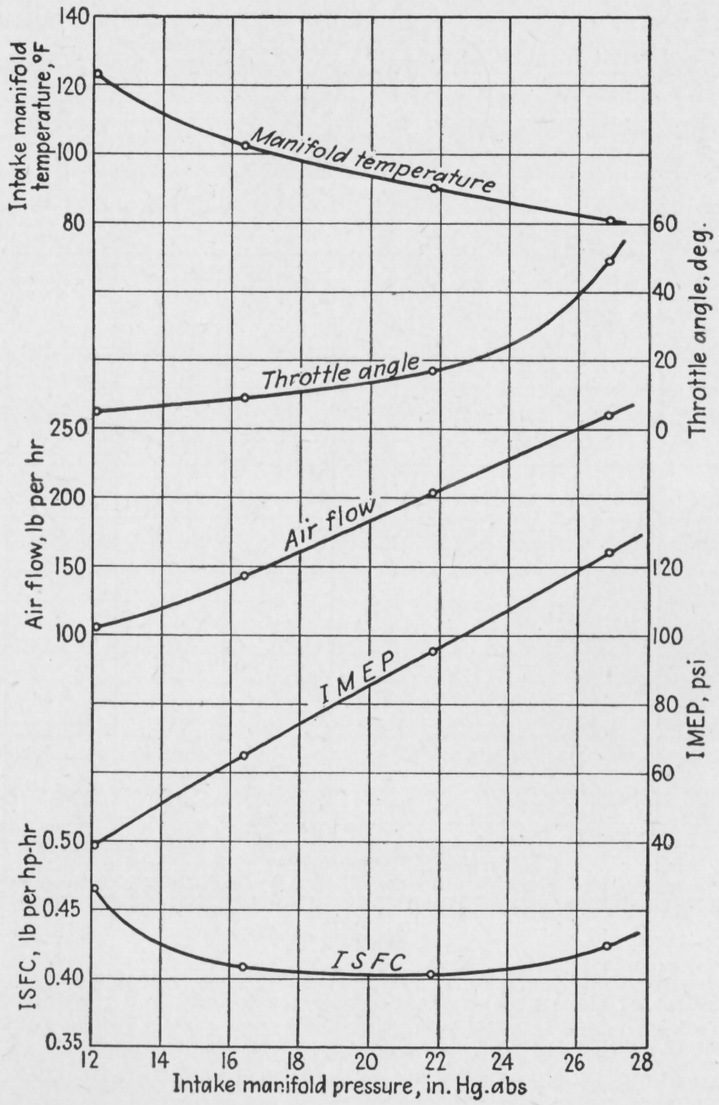
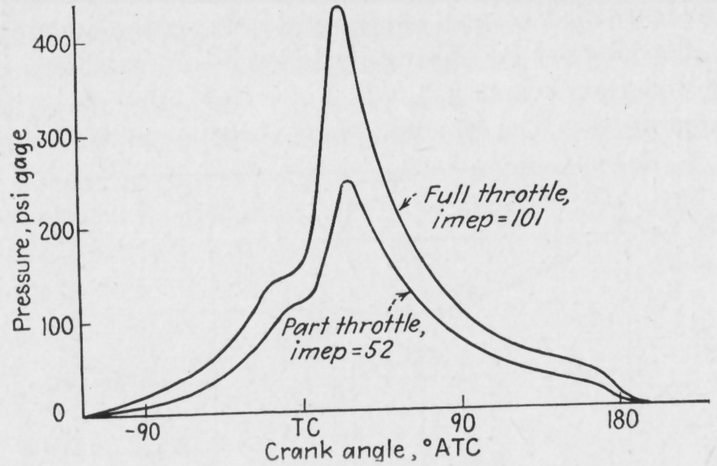
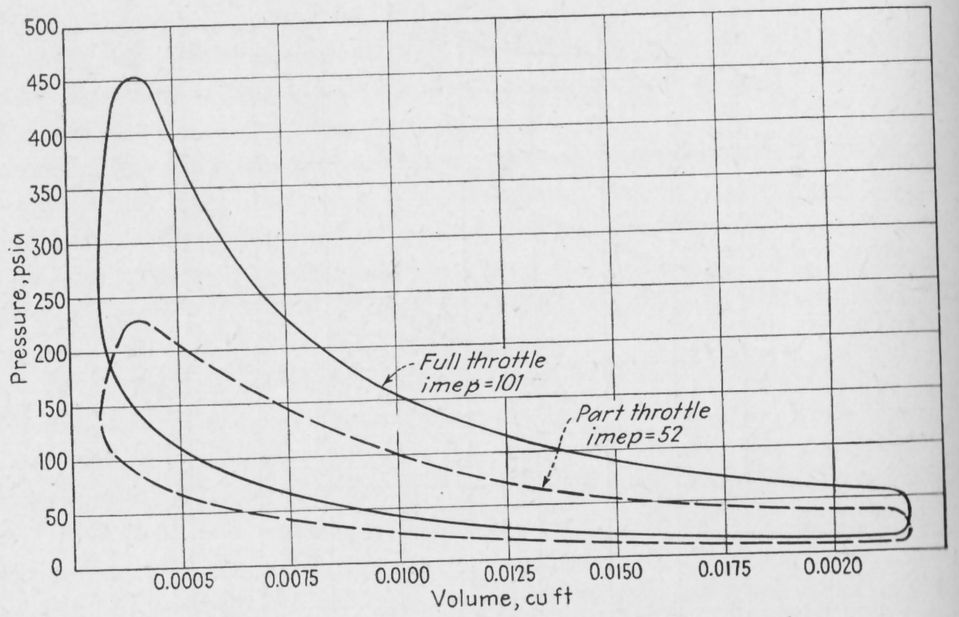


FIG. 83. Effects of throttling on a six-cylinder Plymouth passenger-car engine. (*Case Institute of Technology* at 1,500 rpm.)

The combined effects of throttling on the imep, isfc, and air flow to the cylinders of a passenger-car engine are shown in Fig. 83 as a function of the absolute pressure in the intake manifold. As a matter of interest, curves for the charge temperature in the intake manifold and the angular setting of the butterfly throttle valve are also given. To



(*a*)



(*b*)

FIG. 84. Effect of throttling on the pressure–crank-angle and pressure-volume diagrams for a six-cylinder Plymouth engine at 1,500 rpm. (*Case Institute of Technology.*)

show the effects of throttling on the indicator diagram, indicator diagrams for the one-half and full-load conditions are given in Fig. 84 for the same test conditions. So far as the imep and isfc are concerned, the effects of a loss in volumetric efficiency are essentially the same as those of throttling.

In two-stroke-cycle spark-ignition engines the exhaust gas can be scavenged from the cylinder only by forcing the fresh charge in under pressure. The change in charge density that can be effected by

reducing the pressure in the cylinder at the beginning of the compression stroke is almost negligible—the energy released in the cylinder each cycle must be controlled almost entirely by the extent to which the fresh charge is diluted by the exhaust gas. Thus the exhaust-gas concentration at light loads is quite high, making it necessary to use

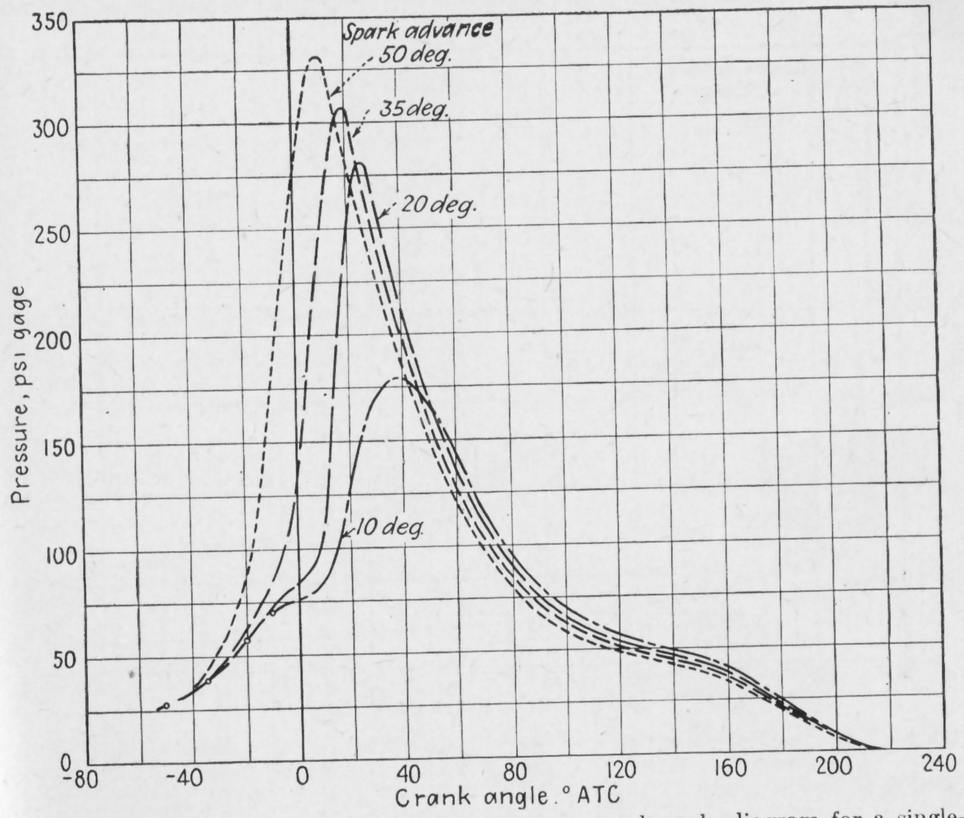


FIG. 85. Effect of spark advance on the pressure–crank-angle diagram for a single-cylinder CFR engine. Compression ratio = 5.0:1, rpm = 1,200, and fuel-air ratio = 0.075. (*Case Institute of Technology.*)

very rich mixtures to obtain reasonably satisfactory combustion in that range.

It is interesting to compare the effects of load reduction in four-cycle spark-ignition engines with those in diesel engines. Figures 80, 81, 83, and 84 are helpful in this connection. The exhaust temperature in the diesel increases practically linearly with imep in the normal operating range. In spark-ignition engines it remains substantially constant for the ideal cycle, although it does increase for the actual cycle because the heat losses become proportionately lower as the quantity of charge per cycle is increased. The air flow increases practically linearly with load in the spark-ignition engine, whereas it

remains substantially constant in the diesel. In fact, the air flow falls off somewhat with an increase in load in the diesel because of higher combustion-chamber temperatures and hence lower volumetric efficiencies at the higher loads. In both types of engine, the peak pressure increases with load by about the same percentage, but the pressures are higher in the diesel at corresponding imeps. This is

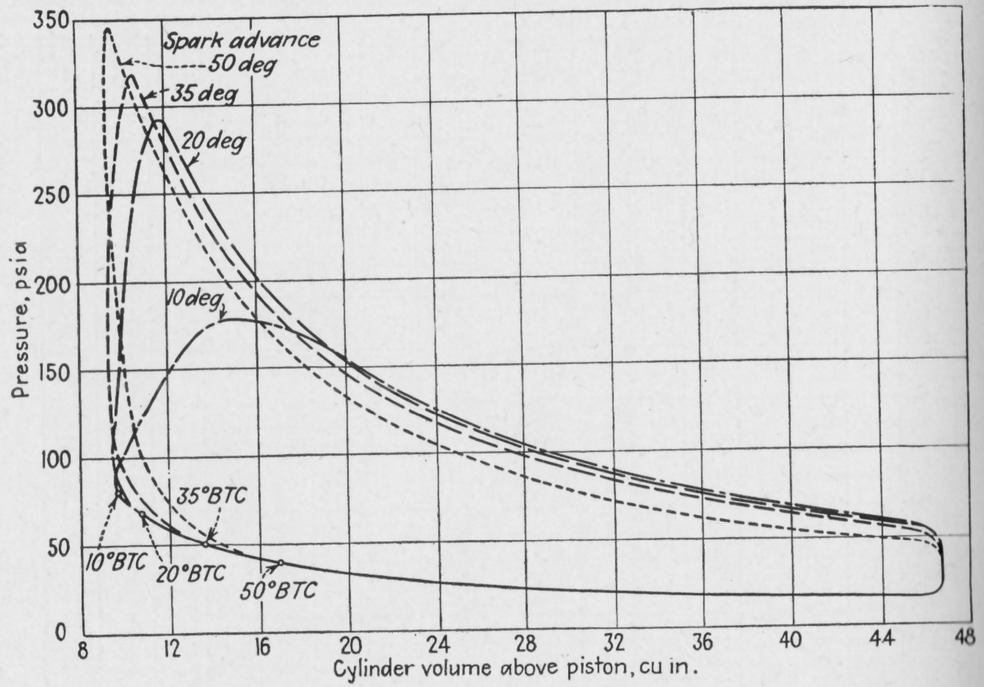


Fig. 86. Effect of spark advance on the indicator diagram for a single-cylinder CFR engine. Compression ratio = 5.0:1, rpm = 1,200, and fuel-air ratio = 0.075. (*Case Institute of Technology.*)

brought about by the higher compression ratios. Note, too, that the diesel indicator diagrams show that the pressures during the compression stroke are unaffected by a change in load, whereas the pressure at any given point during the compression stroke of the spark-ignition engine falls off as the load is reduced.

Effects of Spark Timing in Spark-ignition Engines. The effects of spark timing on the *P-T* and *P-V* diagrams are shown in Figs. 85 and 86. Note the manner in which too early a spark advance causes an excessive loss in area at the lower left corner of the diagram, while too late an advance causes a similar loss in the region at the upper left end. While the quantitative values for the effects of spark advance on power and economy vary widely from one engine to another, the curves in Fig. 87 drawn from data taken at the same

time as the diagrams in Fig. 86 are characteristic of the response of most engines. As mentioned previously, all points on these curves are for detonation-free operation. The effects of spark advance on detonation are discussed in other chapters.

An indication of the importance of the interrelationships between engine operating variables is indicated by Fig. 88, which shows mix-

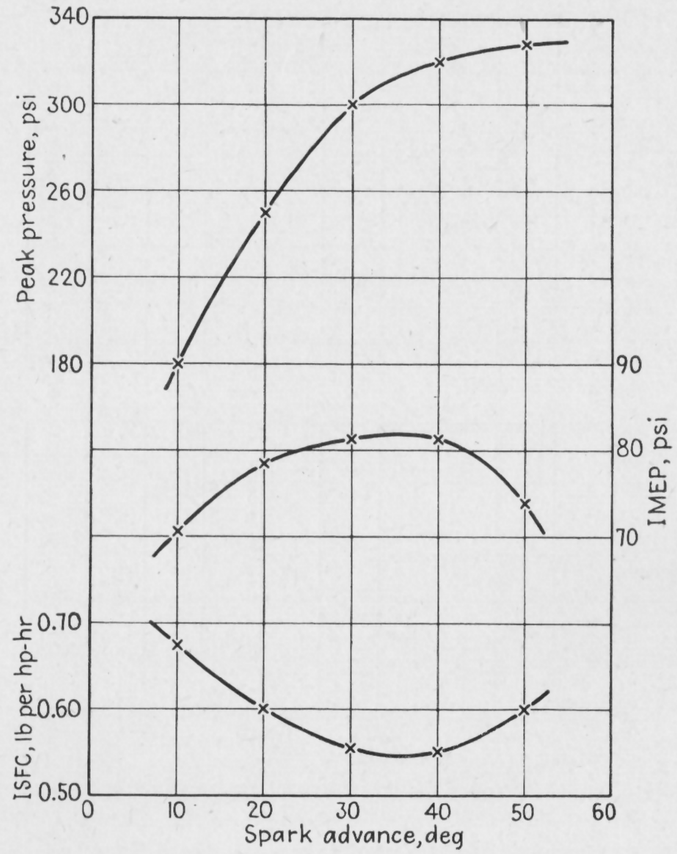


Fig. 87. Effect of spark advance on imep, isfc, and peak pressure. The data are from the same test as that of Fig. 86. (*Case Institute of Technology.*)

ture loops for a series of different spark advances. These curves clearly show that any set of data taken at a given spark advance to determine the optimum fuel-air ratio is good only for that spark advance. Conversely, any set of data taken at a given fuel-air ratio (or, worse, an uncontrolled fuel-air ratio) is good only for the fuel-air ratio of the test and should not be compared directly with data for a different fuel-air ratio. This shows one of the reasons why all operating variables should be carefully and systematically controlled in any test work.

Effect of Injection Rate and Advance in Diesel Engines. As pointed out in the previous chapter and as shown in Fig. 62, the injection advance affects both the timing of the beginning of combustion and the combustion rate. The resulting effects on engine performance are indicated in Fig. 89. Variations in fuel-spray characteristics (*i.e.*, droplet size and dispersion, etc.) and the rate of injection

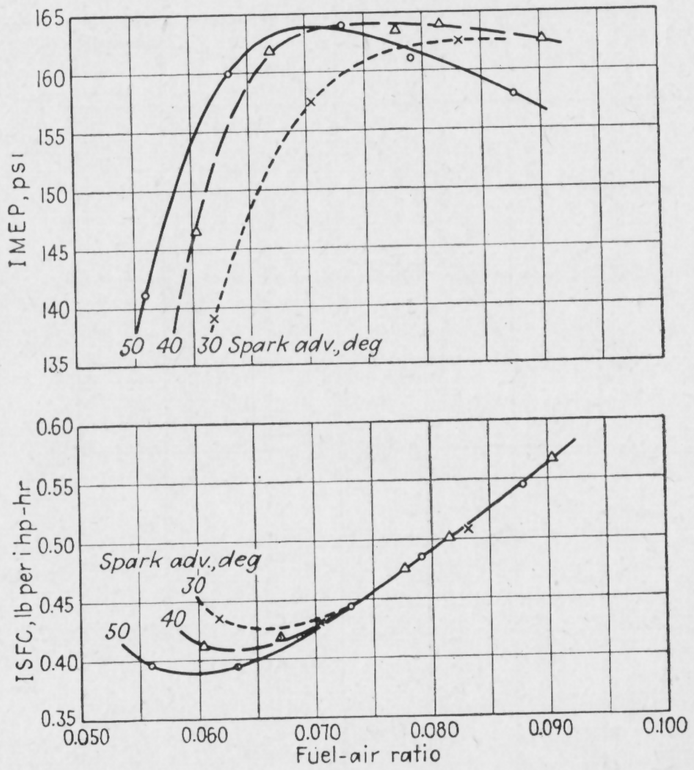


FIG. 88. Effect of spark advance on the shape of the mixture loop for a 1,500-hp aircraft engine.

also have pronounced effects on the rate of combustion. The relationships between these factors are so complex that much experimental work must be carried out before a satisfactory combustion rate can be achieved.

Valve Timing. The design of a cam and the timing of the valve opening and closing events present many interrelated problems. From the standpoint of the indicator diagram the most important single factor is the speed range for which an engine is intended. The volume of gas that must flow through the restriction formed by the valve and the cylinder port each cycle is constant. The flow rate at any given instant depends on the valve opening area and the pressure

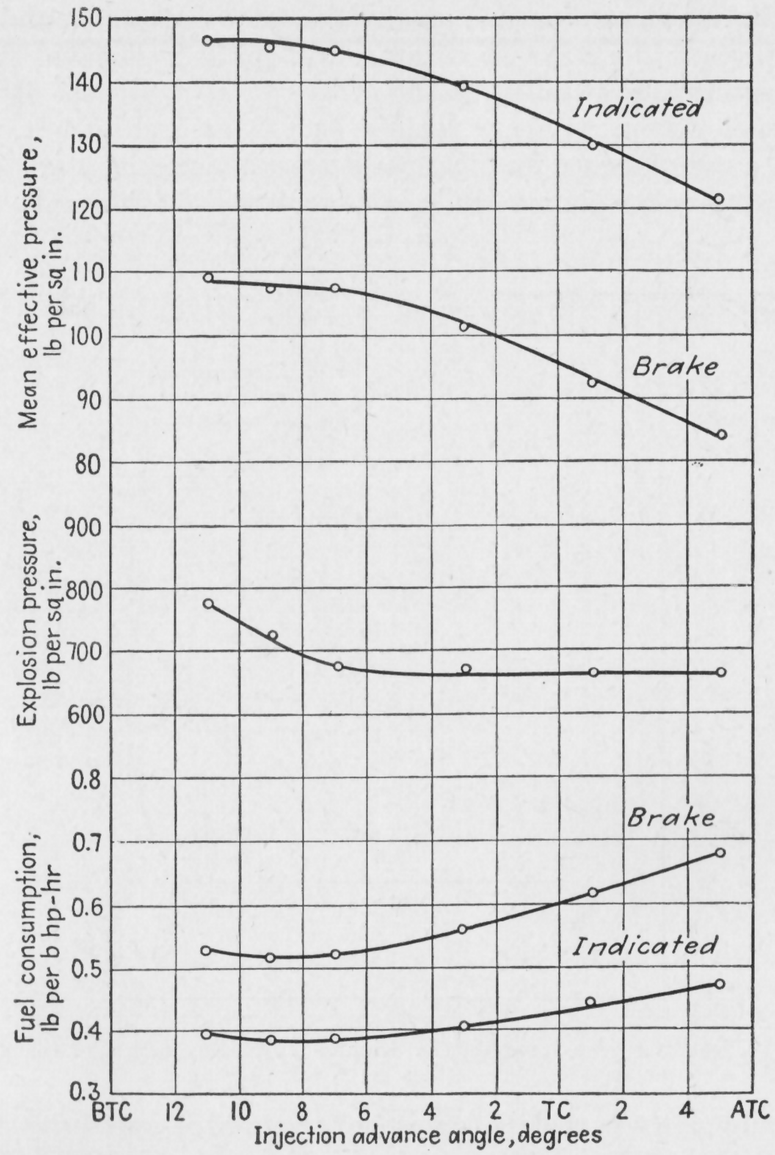


FIG. 89. Effect of injection advance angle on engine performance. Engine speed, 1,500 rpm; fuel consumption = 0.0003 lb per cycle, 12 per cent excess air, compression ratio = 13.5:1. (*Rothrock and Waldron, NACA Tech. Rept.* 525, 1935.)

differential between the cylinder and the manifold. For example, in the case of the exhaust blowdown event, if the exhaust valve were opened instantaneously, it would take a certain length of time for the blowdown to occur. To have the least loss in indicator-card area, analysis discloses that the valve timing should be set so that about half of the blowdown occurs before and half after bottom center (see Fig. 84). Since the time required for the blowdown in this idealized

case is independent of engine speed, it follows that a cam and valve timing intended for a low speed range will not be well suited to high-speed performance because, among other things, it will not allow the exhaust blowdown to occur rapidly enough at high speeds. Conversely, a cam designed for high speeds must of necessity start to open the exhaust valve quite early. As a result the "high-speed" profile

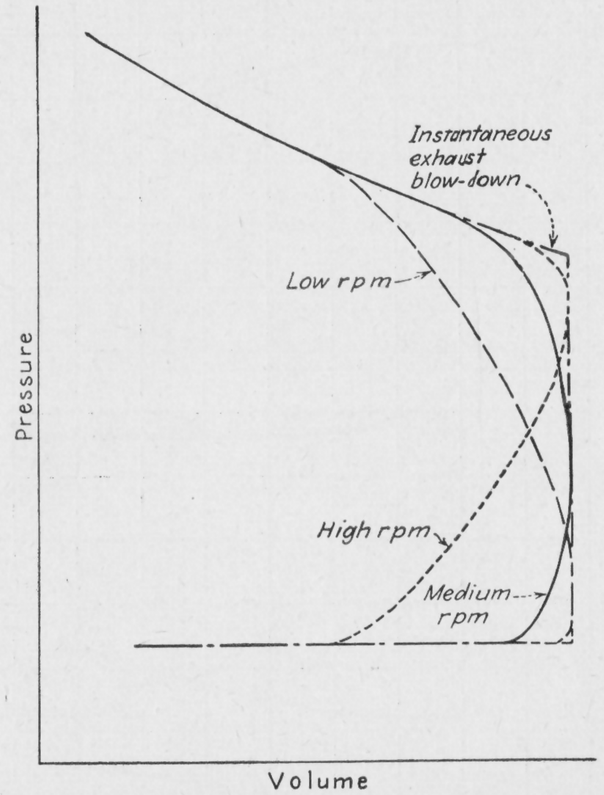


FIG. 90. Effect of engine speed on the exhaust blowdown portion of the indicator diagram for a representative engine with a given valve timing.

cam would give a considerable loss in power at low speeds by cutting off the "toe" of the indicator card, as indicated in Fig. 90. Actually the relations are more complex than this brief survey indicates, and the intake event must be considered as well as the exhaust. However, although one might expect the over-all actual effects of rpm to be quite large, it has been found that they are not if the speed range is not too great. In general, engines are usually much less sensitive to valve timing than to spark timing.

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Problems

1. Define the following terms:

bhp	Volumetric efficiency
ihp	Exhaust blowdown
fhp	Valve overlap
bmep	Scavenging
imep	Residual exhaust gas
fmep	Clearance volume
bsfc	Best-power mixture
isfc	Best-economy mixture

2. Show by means of superimposed sketched diagrams the difference between a cycle in an actual spark-ignition engine and its idealized constant-volume counterpart. Consider the latter to have been computed with Hottel charts for a set of conditions giving the same weight of fresh charge per cycle as in the actual engine. Use three types of diagram, viz., pressure–crank-angle, linear P-V, and log-log P-V.

3. Given the indicator-diagram data in the table below, plot pressure–crank-angle, log-log P-V, and linear P-V diagrams. Measure the area of the latter, and compute the thermal efficiency, isfc, and imep from the area. Also compute the volumetric efficiency, isfc, and imep from the test data at the top of the table, and compare the values for the imep and isfc computed by the two methods. (*Note:* The induction system on this engine had been intentionally restricted to give an unusually low volumetric efficiency.)

INDICATOR-DIAGRAM DATA

(Obtained from a Single-cylinder CFR Engine with a Balanced-pressure-valve Pickup)

Piston displacement	= 37.4 in.^{3}	Spark advance	= 30 deg
Compression ratio	= 5.0:1	Rpm	= 1,200
Fuel flow	= 2.46 lb/hr	Bhp	= 2.49
Air flow	= 32.6 lb/hr	Fhp	= 1.96
Barometric pressure	= 29.48 in. Hg	Room temperature	= 69°F

Crank angle, °ATC	Cylinder pressure, psi (gauge)	Cylinder volume, in.^{3}	Crank angle, °ATC	Cylinder pressure, psi (gauge)	Cylinder volume, in.^{3}
−180	−2.1	46.75	27	250	11.9
− 51	25	17.8	37	220	14.0
− 24	50	11.3	48	170	16.8
− 12	75	9.7	63	125	21.4
0	105	9.35	70	105	23.7
5	125	9.40	90	75	30.0
8	170	9.45	128	50	40.5
12	220	9.6	154	42	45.2
15	250	9.9	180	20	46.75
20	300	10.7	200	5	45.7

4. Using the Hottel charts, calculate the thermal efficiency, imep, and isfc for an idealized cycle indicating the possibilities of the cycle in Prob. 3 if there were no reduction in charge density caused by either a pressure drop through the induction system or a temperature rise through heat transfer from the combustion-chamber walls. Use room pressure and temperature except for the effects of dilution of the fresh charge by the residual exhaust gas in the clearance volume.

5. Repeat Prob. 4, using the same conditions except for the pressure and temperature at the beginning of the compression stroke. Use the initial pressure of −2.1 psi gauge from the indicator-diagram data. Allow for heat transfer from the cylinder walls to the fresh charge by computing the initial temperature from the volume of the cylinder and the weight of charge actually induced each cycle as defined by the air flow. Assume that the amount of residual-exhaust-gas dilution is the same as in Prob. 4.

6. Superimpose plots of data from Probs. 4 and 5 on the linear P-V diagram of Prob. 3. Tabulate data for the thermal efficiency, volumetric efficiency, isfc, and imep for the three cases. Compare the data, and discuss briefly.

7. Determine the area of the portion of the linear P-V diagram for Prob. 5 that lies in the pressure region above a pressure equal to 75 per cent of the peak pressure. Express this area as a percentage of the area of the complete diagram. Discuss briefly the relation between the losses caused by reducing the peak pressure in this way and the percentage reduction in peak pressure effected.

8. Demonstrate the effects of throttling and supercharging by repeating the calculations carried out in Prob. 4, using the same set of conditions except for the inlet pressure. Determine the thermal efficiency, imep, and isfc for an intake-pipe pressure of (*a*) 7.35 psia and (*b*) 29.4 psia.

9. Demonstrate the effects of charge inlet temperature by repeating the calculations of Prob. 8 for an intake-pipe pressure of 14.7 psia and an intake-pipe temperature of (*a*) −20°F and (*b*) 200°F. Tabulate and compare the values from Probs. 4, 8, and 9.

10. What effect does throttling have on a spark-ignition-engine cycle from the standpoint of (*a*) thermal efficiency of ideal cycle, (*b*) peak pressure of actual cycle, (*c*) imep of actual cycle, (*d*) isfc of actual cycle, and (*e*) per cent residual exhaust.

State answers in terms of the effect of progressively *increasing* the intake manifold absolute pressure, *i.e.*, reducing the amount of throttling.

11. Engine flywheels are often marked to facilitate the checking of valve and spark timing. The marks are usually scribed only for the No. 1 cylinder. Indicate with a sketch the relative position of the following markings found on an engine flywheel: TC; IO; IC; EO; EC. Estimate angles on the basis of common values for automotive engines.

12. If the steepest portion of the exhaust-blowdown part of the pressure–crank-angle diagram occurs 54 deg after the exhaust valve starts to open, determine the exhaust-valve closing timing for best performance if the exhaust cam is designed to keep the exhaust valve open for 240 deg.

13. Consider the following three forms of combustion chamber: (*a*) a 4-in. sphere with a spark plug at the center; (*b*) a 4-in.-diameter cylinder 8 in. long with a spark plug in the center of one base; (*c*) a 4-in.-diameter cone 8 in. long with a spark plug in the center of the base.

Estimate (within ±5 per cent) the flame-front area for the following distances of the flame from its point of origin: 0, 2, 4, 6, and 8 in. Plot superimposed curves for the flame-front area vs. flame travel for the three cases. State in which instance the highest combustion rate would prevail, and indicate which form of combustion-rate curve would be best suited to an internal-combustion engine.

14. With the help of sketches of P-V diagrams, indicate briefly the effects on a spark-ignition engine cycle of the following: (*a*) exhaust-valve timing; (*b*) spark advance; (*c*) use of one vs. two spark plugs at opposite ends of the combustion chamber.

15. With the help of sketches of P-V diagrams, indicate briefly the effects on the cycle of a high-speed compression-ignition engine of the following: (*a*) reduction of the amount of fuel injected each cycle; (*b*) injection advance; (*c*) rpm.

16. Estimate as accurately as possible the ihp, bhp, isfc, and bsfc of a 300-in.^{3}-displacement 7.0:1-compression-ratio four-cycle spark-ignition engine operated at 1,600 rpm with a fuel-air ratio of 0.067 using commercial gasoline. Assume representative values for the volumetric efficiency, the cycle-efficiency ratio, and the mechanical efficiency. *State assumed values clearly.*

17. Suppose that you are approached by a man who claims to have a dope for gasoline that will permit an increase in compression ratio from 6.5:1 to 8.0:1. He further claims that such an increase will give 25 per cent more power and 25 per cent less fuel consumption. Make your own estimate of the maximum possible improvements obtainable from such a change in compression ratio.

CHAPTER VI

FUEL METERING AND INJECTION

One of the most difficult single sets of problems associated with the practical development and production of any engine is that concerned with the metering of the fuel. The fuel-metering device determines to a large degree the fuel economy, smoothness of operation, flexibility, and acceleration characteristics of an engine. Most of the problems of fuel metering are so intimately bound up with the characteristics and limitations of the particular metering device that the first topic cannot very well be presented apart from the second. For this reason in this chapter the section devoted to the operating principles of each type of fuel-metering equipment is followed by a section giving a brief description of a representative commercial unit that illustrates the application of these principles.

MIXTURE REQUIREMENTS OF SPARK-IGNITION ENGINES

Effect of Air Flow on the Required Fuel-Air Ratio. The close relation between engine power output and the rate of air flow to the cylinders was demonstrated in Chap. V. It happens that air flow is the most important variable upon which engine mixture requirements depend. Ideal-cycle considerations indicate that, in spark-ignition engines, the fuel-air ratio should be close to the chemically correct value. Certainly from consideration of the combustibility limits shown in Fig. 45 it is evident that the fuel-air ratio should not exceed 0.125 or be less than 0.052. These general observations are basically correct, but—as in most engine problems—certain modifications of the general theory must be made for best service performance.

The mixture loop shown in Fig. 82 and the accompanying discussion showed that there is both a best-economy and a best-power mixture at any given engine speed and throttle opening. The most important single variable determining the fuel-air ratios for each of these conditions is the amount of residual exhaust gas, for that quantity increases rapidly as the engine is throttled to an idle and materially slows down the flame speed. It also reduces what might be referred to as the *effective fuel-air ratio* in the cylinder at the time of

the spark, for dilution of the fresh charge by exhaust gas reduces the relative proportion of combustible vapor. Because of this, the best-power and best-economy mixtures under idling and low power-output conditions are considerably richer than at moderate or high power outputs. Figure 91 shows this characteristic very nicely in the form of curves for the best-power and best-economy fuel-air ratios for a representative passenger-car engine. Note that these curves closely parallel the chemically correct fuel-air ratio except at the idle end.

The best-power fuel-air ratio defined by Fig. 82 for the indicated output of a single-cylinder engine is somewhat richer than the chemi-

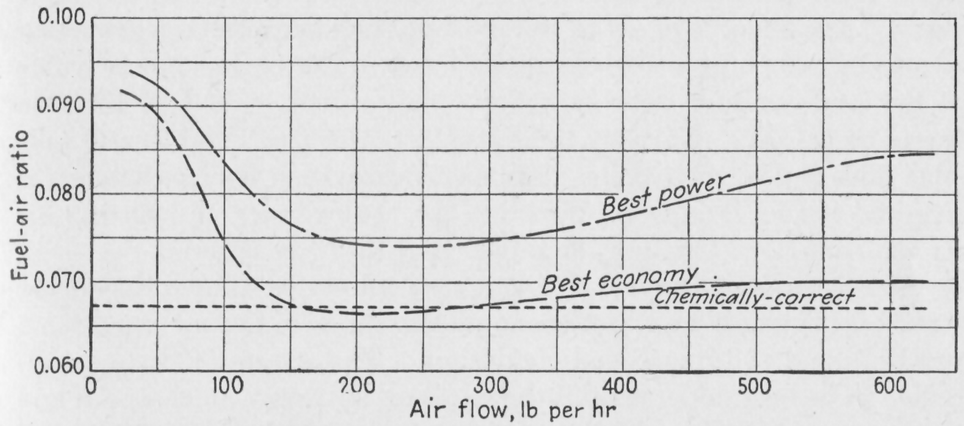


FIG. 91. The best power and best economy mixture requirements of a representative passenger-car engine shown for comparison with the chemically correct fuel-air ratio. All curves are for normal road-load operating conditions. (*Case Institute of Technology.*)

cally correct value. In a multicylinder engine the best-power fuel-air ratio occurs at an even richer fuel-air ratio because of the effects of uneven distribution of the fuel to the various cylinders. It also happens that better acceleration characteristics are obtained with mixtures considerably richer than the chemically correct value. Both these factors are discussed in Chap. IX. Aside from power considerations, it is also true that some engines require rich mixtures at high power outputs either to inhibit detonation or to reduce the exhaust temperature and thus avoid overheating and possible warping of the valves. The loss in fuel economy attending mixture enrichment together with the extra complication required to achieve it has limited the use of "power enrichment" in automotive engines. Mixture enrichment at high outputs has, however, played an extremely important part in aircraft engines where the extra complication and expense can be justified by the increase in permissible take-off power output.

Special Provisions for Acceleration. Some sort of special device is incorporated in most carburetors to give richer mixtures during acceleration and thus increase the rate at which the engine will respond to a sudden opening of the throttle. Mixture enrichment for acceleration is needed because the less volatile portions of the fuel tend to accumulate on the intake manifold walls and flow relatively slowly in liquid form along the walls toward the intake ports. An abrupt opening of the throttle immediately increases the flow rate of the air and vaporized fuel, but an appreciable period of time must elapse before the heavier components of the increased fuel flow can travel in liquid form along the manifold walls from the carburetor to the valve ports. The effect is likely to be especially bad because the carburetor is usually set to give fuel-air ratios close to the best-economy value at low and medium outputs. Any small reduction in fuel-air ratio therefore causes a relatively large loss in power (see Fig. 82) and possible misfiring of the engine. No special provision for acceleration is required if fuel is injected directly into the cylinder or into the air stream as it flows through the intake port into the cylinder.

The Road-load Curve. While it was not stated above, the curves for the best-power and best-economy mixtures in Fig. 91 were for a particular set of typical load conditions. The nature of these conditions can be best understood if the relations between vehicle speed and the power required to drive a vehicle at a constant velocity on level road are considered. Figure 92 shows a curve of this sort for a typical passenger car. This curve, called a *road-load curve*, shows that the normal load imposed on the engine by the vehicle is small at low speeds but increases slowly at first and then at a progressively rapid rate as the speed is increased until the full-throttle output of the engine is required. This point defines the top speed of the vehicle. It is true that heavy loads may be imposed on an engine at low speeds during acceleration or hill climbing, while light loads at high speeds are found during deceleration or on a downgrade, so that these conditions must also be considered. However, since engine operation under conditions that are appreciably off the road-load curve constitutes a minor portion of passenger-car operation, the problem is greatly simplified if the behavior of a fuel-metering device is considered primarily from the standpoint of a road-load curve. The effect of deviations from the road-load curve on the metering characteristics of the device can then be considered separately.

Engines used in trucks, busses, tractors, etc., operate at loads at or near the full-throttle value for a much greater proportion of their

service life than passenger-car engines. Thus, for such vehicles, the full-throttle curve as a function of speed is more important than the road-load curve. The curve for the full-throttle engine output is included with that for the road-load power requirements in Fig. 92. Most service operation lies somewhere in the region between the two curves.

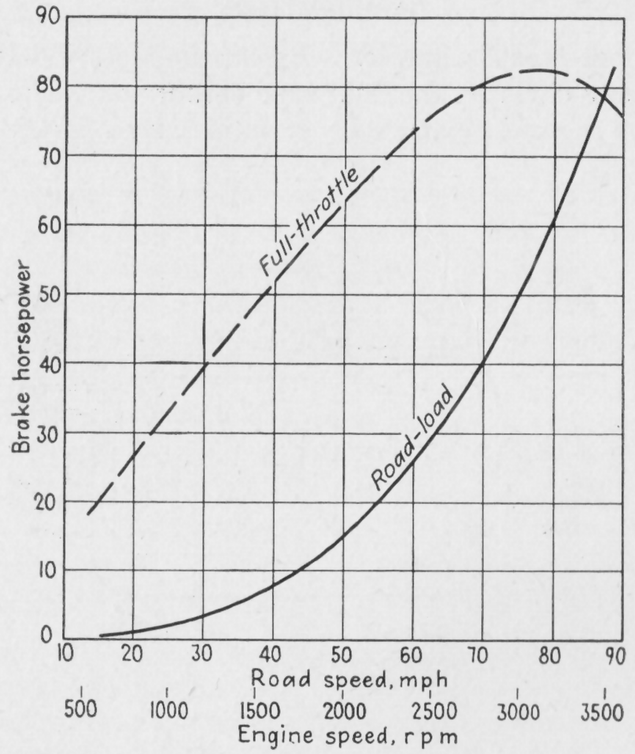


FIG. 92. Full-throttle and road-load horsepower as a function of speed for a passenger-car engine operated in high gear.

The road-load curve can be approximated by a semicubical parabola; *i.e.*, the power varies very nearly as the cube of the speed. This follows from the fact that the chassis-friction and tire-rolling resistance vary with speed at a rate somewhat greater than the linear value, while the air resistance of the vehicle varies as the square of the speed. Thus the power—*i.e.*, the product of the resistance and the speed—varies approximately as the square of the speed at low speeds and very nearly as the cube of the speed at high speeds, where the air resistance is the most important factor. This matter will be discussed in more detail in Chap. XIII, Performance Analysis.

Load Compensation. The effect on a carburetor of load variations at constant speed is often important. A fuel-metering device is said

to have good *load compensation* if its characteristic curve of fuel-air ratio vs. air flow is practically unaffected by load variations at any given engine speed. Good load compensation is sometimes difficult to get. As a matter of fact, inability to obtain it has been the principal shortcoming of many patented metering devices that have proved unsuccessful.

CARBURETORS

Simple Float-type Carburetor. By far the most widely used type of fuel-metering device is the float-type carburetor. The diagram in Fig. 93 shows the most elementary form of such a device. The butterfly throttle valve shown is usually incorporated in the carburetor although it is generally not a functional part of the main metering system. The float chamber and float valve are employed to maintain the fuel at a level such that fuel almost overflows out through the fuel-discharge nozzle into the venturi. In effect, this maintains the fuel pressure ahead of the fuel-metering jet equal to the air pressure ahead of the venturi. If air flows through the venturi, a reduction in static pressure in the venturi throat results. This induces fuel to flow through the fuel-metering jet at a rate that depends on the air flow.

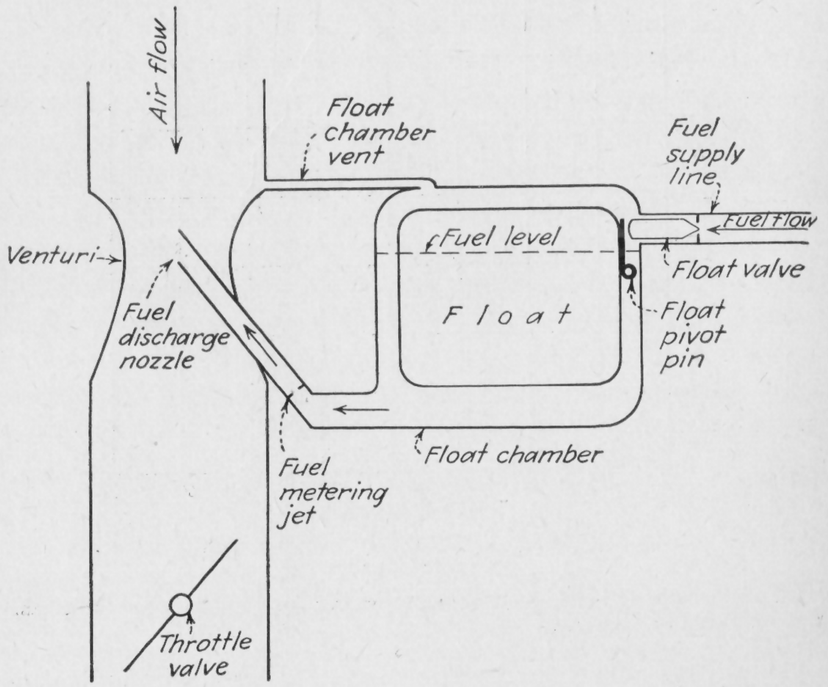


FIG. 93. Schematic diagram of a simple float-type carburetor.

The principles of operation of these units are quite simple; the venturi and fuel-metering jet operate in parallel in such a way that the pressure differential between the venturi inlet and throat is equal to the pressure drop across the fuel jet. Since these pressure differentials are directly proportional to the square of the flow through their respective restrictions and since the pressure differentials are maintained equal, a constant fuel-air ratio should result.

The pressure-drop–flow relations for a carburetor may be demonstrated more explicitly by making use of the fundamental relations

$$V = CA\sqrt{2gH} \qquad \text{and} \qquad W = wV = wCA\sqrt{2gH}$$

where V = volume of fluid flowing, ft³/sec
W = weight of fluid flowing, lb/sec
w = density of fluid flowing, lb/ft³
H = differential in pressure head between the inlet to and the throat of the orifice, ft of the fluid flowing
A = area of the orifice throat, ft²
C = orifice coefficient

Using the subscripts a and f to denote air and fuel, respectively, we may write

$$W_a = w_a V_a = w_a C_a A_a \sqrt{2gH_a}$$

and

$$W_f = w_f V_f = w_f C_f A_f \sqrt{2gH_f}$$

In a float-type carburetor the fuel level in the float chamber is maintained at approximately the same level as the fuel-discharge nozzle so that the pressure drop across the fuel jet may be considered as equal to the pressure differential between the inlet and the throat of the air venturi. Since the fuel-metering jet is in the form of a small, sharp-edged orifice in a relatively large passage, virtually all the velocity energy at the throat is dissipated in eddies below the orifice. Thus the over-all pressure drop across this jet is essentially equal to the pressure differential between its inlet and its throat, and hence

$$H_a w_a = H_f w_f \qquad H_a = H_f \frac{w_f}{w_a}$$

The fuel-air ratio thus becomes

$$\frac{F}{A} = \frac{w_f C_f A_f}{w_a C_a A_a}\sqrt{\frac{2gH_f}{2gH_a}} = \frac{w_f C_f A_f}{w_a C_a A_a}\sqrt{\frac{H_f w_a}{H_f w_f}} = \frac{C_f A_f}{C_a A_a}\sqrt{\frac{w_f}{w_a}} \qquad (10)$$

The above simplified relation is a first approximation that holds only for air flow rates at which the density of the air in the venturi throat

is not reduced appreciably below the value at the venturi inlet, *i.e.*, in the range where the pressure differential between the inlet and the throat of the air venturi is small.

The fuel-air ratio furnished by a simple float-type carburetor has been found in practice to vary appreciably with air flow because actual units differ from the assumed idealized case.[1] One of the most important differences is that, since fuel should not dribble out through the

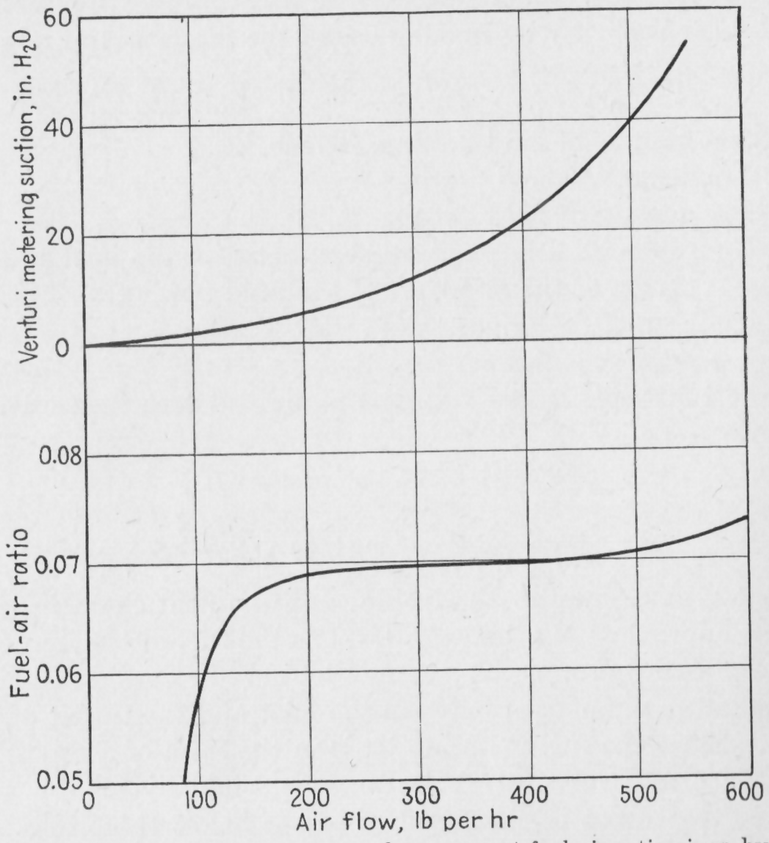


FIG. 94. The venturi metering suction and consequent fuel-air ratio given by a simple float-type carburetor.

discharge nozzle when the engine is stopped even if the vehicle should be on a slope, the float must be adjusted to maintain a fuel level appreciably below the discharge-nozzle outlet. As a result, no fuel will begin to flow until the air flow, and hence the air-venturi suction, becomes sufficiently high to draw fuel up to the level of the discharge opening. Even then at low fuel-flow rates the surface tension of the fuel exerts a force restraining fuel flow. Another important factor causing the fuel-air ratio to be affected by air flow is the compressibility of the air. At high air flow rates the static pressure in the

throat of the venturi drops sufficiently to cause an appreciable reduction in air density. As a result, although the velocity ratio for the flows through the air venturi and the fuel jet may remain constant, the fuel-air ratio does not, for it is a weight ratio. Thus the reduction in air density at the throat of the venturi at high air flows causes an increase in the fuel-air ratio.

Figure 94 shows curves for both the *venturi suction*, or *metering head*, and the fuel-air ratio given by a simple float-type carburetor. The usable portion of the curve is evidently that giving venturi metering suctions between about 4 and 50 in. H_2O. Thus in an engine designed for operation over a wide range of speeds the maximum output determines the minimum permissible venturi size (*i.e.*, the power output for which the air flow gives a venturi suction of about 4 in. Hg). It happens that slight variations in fuel level in the float chamber are caused by vibration, off-level road conditions, etc. These, coupled with small random changes in the air-flow pattern through the venturi, make it essential that the fuel-metering head be not less than 4 or 5 in. H_2O if consistently close limits on fuel-air ratio are to be maintained. Thus the lower end of the basic curve for a simple float-type carburetor would not be very usable even if the fuel-air ratio did not fall off rapidly.

Idle Metering System. Many different idle metering arrangements have been worked out to meter at the low-air-flow end of the curve, with varying degrees of success. No idle system has been so satisfactory as the basic main metering system described above. The most successful automotive carburetor idle metering systems have made use of the high-pressure drop across the carburetor throttle valve that occurs when that valve is at or near the closed position. Figure 95 shows a curve for this pressure drop across the carburetor as a whole plotted against rpm for both a road load and the full-throttle condition. Curves for the metering suction, air flow, and the throttle-valve position are also given. Note that Figs. 91, 92, and 95 are all for the same engine and particularly that the carburetor drop is large in the range in which the venturi metering head is small, and vice versa.

An idle system that could be used in conjunction with the main metering system of Fig. 93 is shown in Fig. 96. The passage carrying fuel from the float chamber to the idle discharge nozzle is looped up above the fuel level in the float chamber and vented through an air bleed at the top of the loop to prevent fuel from siphoning through the idle system while the engine is stopped. Thus the suction at the

idle discharge nozzle must be great enough to cause a considerable air flow through the idle air bleed, and thus a pressure drop across that air bleed of at least an inch or two of water if fuel is to be delivered to the engine through the idle system. A glance at Fig. 95 will show

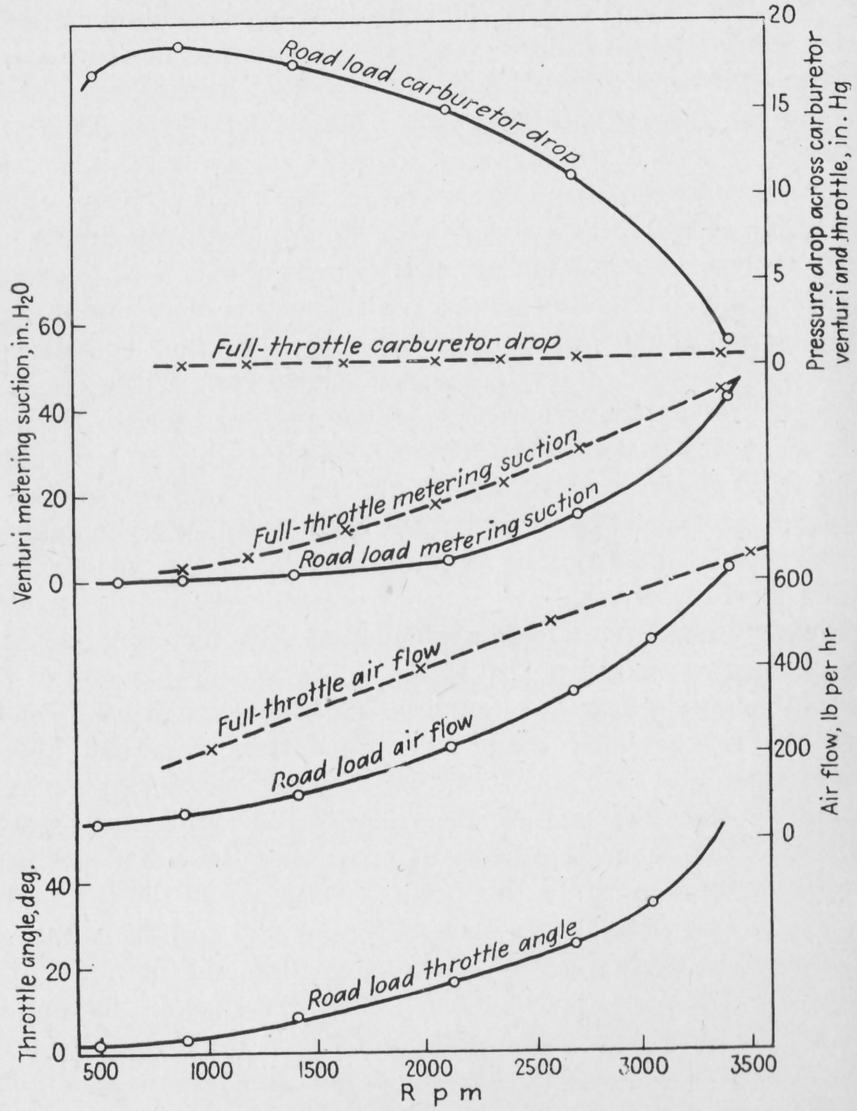


FIG. 95. Effect of speed on factors important in carburetor metering. Dynamometer test-stand data for a six-cylinder Plymouth passenger-car engine. (*Case Institute of Technology.*)

that there is more than ample suction available in the idle range to do this. As the throttle is opened, the suction at the idle discharge nozzle drops off rapidly because the edge of the throttle valve moves away from the idle discharge holes and the static pressure there rises

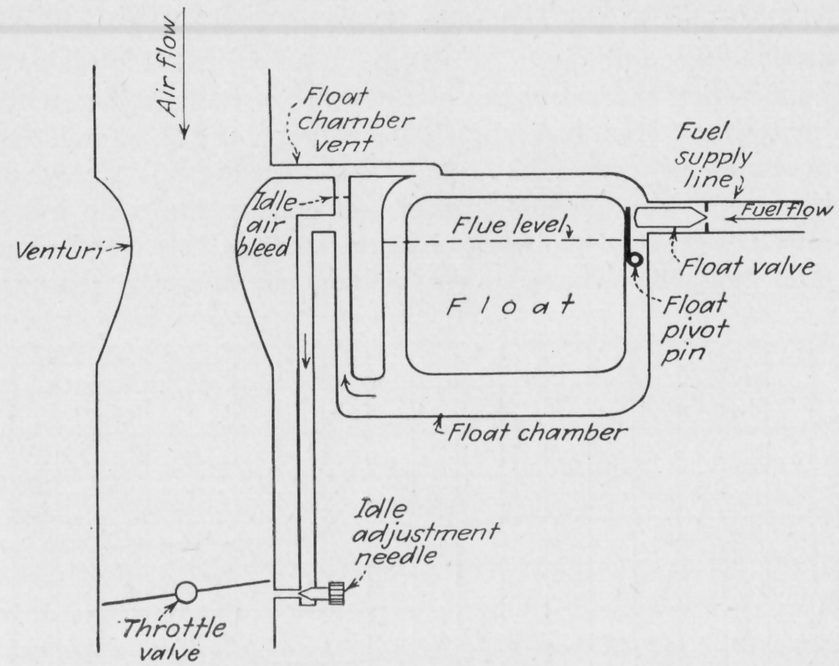


FIG. 96. Schematic diagram for the idle system of a float-type carburetor.

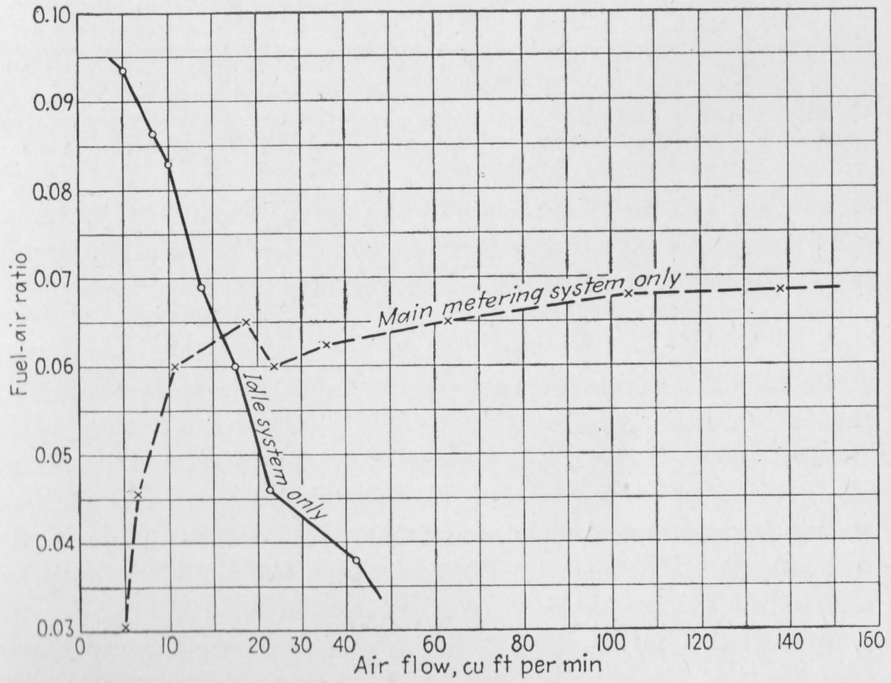


FIG. 97. Metering characteristics of the idle and main metering systems of a float-type carburetor for a passenger-car engine. (*Private communication from J. Meinzinger.*)

toward the value at the carburetor inlet. The idle-system fuel flow rate drops off accordingly.

The metering characteristics of a carburetor built to incorporate a main metering system basically similar to that of Fig. 93 and an idle system similar to that of Fig. 96 are shown in Fig. 97. Two curves are shown—one for the idle system and one for the main metering system. The over-all metering characteristics of this carburetor are shown in Fig. 98. These curves include the effect of the power-

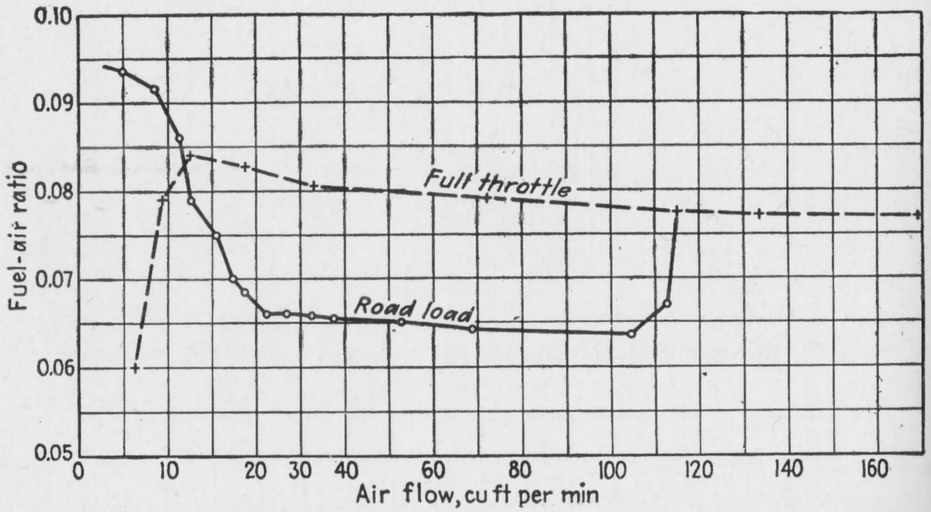


FIG. 98. Road-load and full-throttle metering characteristics of a passenger-car-engine carburetor with a very good setting. (*Private communication from J. Meinzinger.*)

enrichment system and the effects of the operation of each system on the others so that they differ somewhat from a curve obtained by adding the ordinates of the two curves in Fig. 96.

A DOWNDRAFT CARBURETOR OF THE FIXED-JET TYPE

While all carburetors depend on the basic principles outlined in the previous sections, the constructional details of the various makes and models differ widely. One feature of any carburetor, for example, is the direction of air movement through the venturi. While some horizontal-venturi carburetors are in use, the venturi axis is vertical in most models. Those carburetors in which the air flows downward are referred to as *downdraft* carburetors, while those in which the air flows upward are called *updraft* carburetors. Since the air passing through the downdraft units does not have to lift fuel droplets, these carburetors can be made to operate with lower air velocities and hence lower full-throttle pressure drops than would be possible in updraft

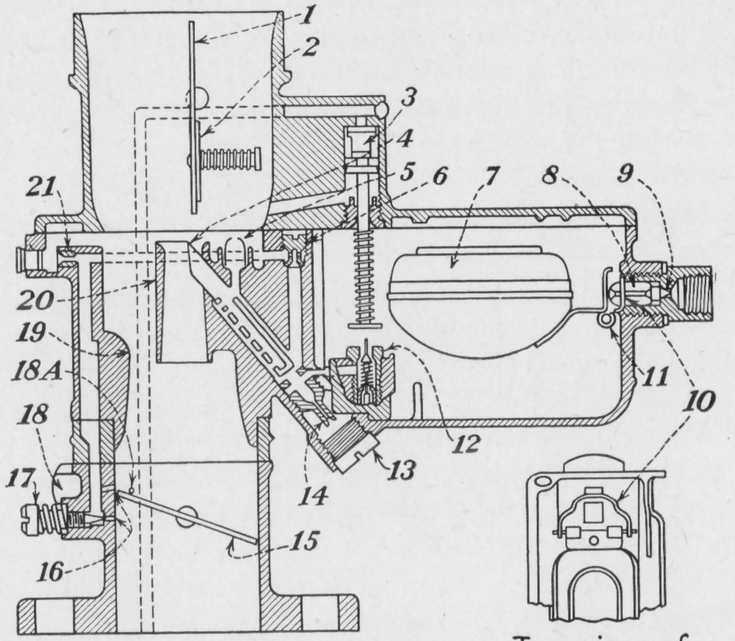
carburetors. Thus they may be designed to give higher full-throttle intake manifold pressures and hence slightly higher power outputs.

The downdraft carburetor shown in Figs. 99 and 100 is representative of the largest single group of fuel-metering devices in use, viz., the float-type, downdraft, fixed-jet, and venturi carburetors. It should be noted that the diagrams in Fig. 99 and 100 are in part schematic in that the relative positions of some of the parts were shifted to simplify the illustration. The float chamber, for example, would lie behind the venturi in Fig. 100 if it were a true view instead of a schematic diagram.

Main Metering System. Fuel enters the carburetor through the inlet connection at the right of Fig. 99 and is admitted to the float chamber by the *float valve.* The float (7) is mounted on a pin (11) so that it controls the float needle valve by the arm extending vertically upward from the pin (11) about which it pivots. The float chamber is vented to the air stream ahead of the venturi through the vent (22) shown in Fig. 100. While it is not definitely shown in Fig. 99, there is a clear passage for fuel to flow from the bottom of the float chamber to the lower, or inlet, side of the main metering jet (14). A screw-type plug (13) is provided below the main jet to make it possible to clean or remove it easily. The fuel flows up through the jet and out the discharge nozzle (4), the metering mechanism being as described in the preceding section. However, the carburetor of Fig. 99 has several additional features in the main metering system that should be mentioned. The first is the *auxiliary,* or *boost,* venturi, placed so that it discharges into the throat of the *primary,* or *main,* venturi. This arrangement nearly doubles the available metering head for a given venturi throat area and hence a given full-throttle air-pressure drop across the carburetor. The second is the *main air bleed* (5), which permits air to enter the fuel as it flows through the passage to the discharge nozzle. Introduction of air bubbles into the fuel in this fashion helps greatly to break up the liquid jet and cause it to form small droplets, which will vaporize quickly.[2]

Idle System. The idle system is also shown in Fig. 99. When the throttle valve (15) is closed to the point where the air flow rate is too low to provide an adequate fuel flow through the main discharge nozzle, the low pressure below the throttle coupled with the high air velocity past the *idle discharge holes* (16) creates a strong suction that draws fuel through the passages leading from the space just above the main jet through the *idle tube* (6) past the *idle air bleed* (21). This, like the main air bleed, serves to help break the fuel up into

small droplets. Even more important is the fact that it prevents fuel from flowing through the idle system except in the idle range where the suction at the idle discharge holes is high. Under such conditions, while air drawn in through the idle air bleed reduces the suction available, there is still enough suction left to lift fuel above the level in the float chamber and carry it out through the idle system. The *idle needle valve* (17) provides a means of adjusting the fuel-air ratio in the idle range. The idling speed of the engine is adjusted with the *throttle stop screw* (37) of Fig. 100. This screw limits the angular travel of the throttle valve, and thus determines the minimum air-flow passage area when the throttle is in the closed position.

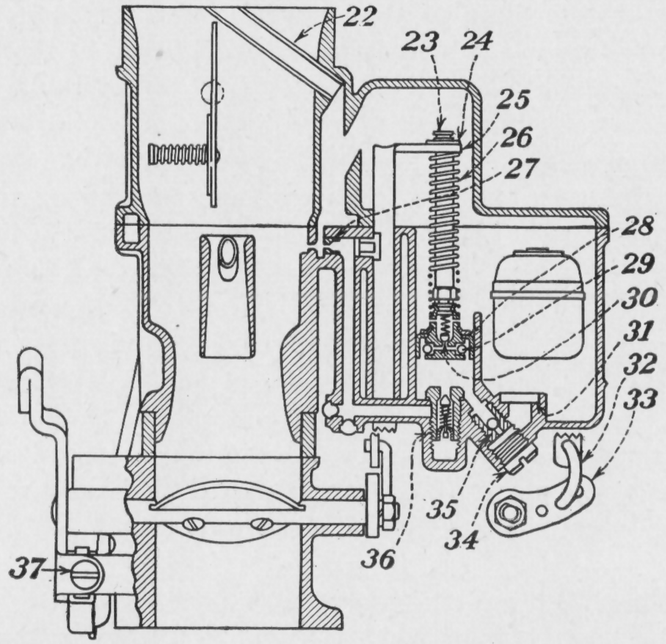


Top view of float mechanism

1. Choke valve
2. Choke relief valve
3. Vacuum piston
4. Main discharge nozzle
5. Main air bleed
6. Idle tube
7. Float
8. Float valve
9. Float valve seat
10. Float fulcrum-pin clip
11. Float fulcrum pin
12. Power-by-pass jet
13. Main jet plug
14. Main metering jet
15. Throttle valve
16. Idle discharge holes
17. Idle needle valve
18. Vacuum spark conn.
18A. Vacuum spark hole
19. Primary venturi
20. Auxiliary venturi
21. Idle air bleed

Fig. 99. Section through a Stromberg fixed-jet carburetor for a Dodge passenger-car engine. The float mechanism, main metering, economizer, and idler systems are shown. (*Bendix Products Division, Bendix Aviation Corp.*)

Choke Valve. The choke valve (1) may be manually operated to give a very rich mixture for starting purposes. This is accomplished by closing off the valve ahead of the venturi but below the float-chamber vent so that the entire space below the choke valve is placed under a suction. This draws fuel through the main metering system at even the very low air flows prevailing during cranking. The *choke relief valve* (2) on the choke valve is a spring-loaded flapper valve that opens if the suction becomes too great. "Choking" is necessary in cold weather because proper combustion in the cylinder cannot occur unless there is a sufficiently rich mixture of vaporized fuel and air. Fuel still in the liquid state at the instant the spark occurs is, of course, of little or no use. At a low temperature of, say, 20°F only the volatile components of a fuel can be evaporated. Therefore under such conditions a combustible mixture can be obtained only by using



22. Float chamber vent
23. Piston connecting rod
24. Cotter pin
25. Pump piston rod
26. Pump duration spring
27. Pump discharge nozzle
28. Pump piston[1]
29. Pump piston expansion spring
30. Pump relief valve
31. Pump inlet strainer
32. Pump link
33. Pump lever
34. Pump inlet plug
35. Pump inlet check valve
36. Pump metering jet
37. Throttle stop screw

Fig. 100. Section through the carburetor of Fig. 99 showing the accelerating pump and the throttle-shaft mechanism. (*Bendix Products Division, Bendix Aviation Corp.*)

fuel-air ratios that are high if figured on a liquid-fuel basis, although the effective fuel-air ratios at the time of ignition on a vapor basis will be no greater than normal. The matter of fuel vaporization under these conditions will be discussed in Chap. VIII.

Many passenger cars are equipped with automatic choke valves operated by thermostats of the bimetal type. Such arrangements often employ a much more complicated mechanism in the carburetor than that described. This is for the purpose of obtaining the proper mixture enrichment under any of a wide range of conditions.

Accelerating Pump. An accelerating pump has been incorporated in the carburetor of Fig. 100 by linking an arm (33) on the throttle shaft to a piston (28) in a cylinder submerged in the float chamber. The *pump piston rod* (25) is not rigidly attached to the *pump connecting rod* (23) but fits loosely around and is free to slide along this rod. Sudden opening of the throttle depresses the spring (26), which acts on the piston (28), forcing it downward at a rate determined by the stiffness of the spring. Fuel that has entered the accelerating-pump cylinder through the *pump-inlet check valve* (35) is forced out through the *accelerating-pump jet* (36), up through drilled passages, and out into the air stream through the *accelerating-pump discharge nozzle* (27).

Economizer. Power enrichment has been provided for in this carburetor in the form of what is called a *power-enrichment, high-speed,* or *economizer* system. The last term, "economizer," was used originally because carburetors were once set for best-power operation at high speeds. Since such a setting gave unnecessarily rich mixtures at part-throttle and low speeds, it seemed proper that a device giving good low-speed economy while maintaining good high-speed performance should be called an *economizer.*

The power-enrichment system shown in Fig. 99 is designed to give mixture enrichment when the intake manifold vacuum drops to about 4.0 in. Hg. The *vacuum power piston* (3) is drawn up against a light spring by the suction transmitted to the upper end of the economizer cylinder through a passage connected to the space below the throttle. As the throttle is opened and the intake manifold vacuum drops, the vacuum power piston is forced downward by the spring until it opens the spring-loaded pintle valve in the *power–by-pass jet* (12). Fuel flows from below this jet through the passage shown in dotted form to the passage leading from the main jet to the main discharge nozzle. Thus the power–by-pass jet meters fuel in parallel with the main jet at or near full throttle.

A VARIABLE FUEL-METERING JET CARBURETOR

Main Metering System. The carburetor shown schematically in Fig. 101 differs from that of the previous section in that a variable rather than a fixed fuel-metering restriction is employed. A metering rod, or pin, is suspended in the main jet. It is so contoured and linked to the throttle that the net area of the passage between the jet and the metering rod is a particular function of the air-flow-passage area where it is restricted at the throttle valve. The float chamber is vented to the air stream at the carburetor inlet, or *air horn,* and the fuel level is maintained by the float in the same fashion as in the carburetor discussed previously. Three venturis in tandem, instead of two, are employed to give an especially high metering head.

The manner in which the main metering system acts to maintain approximately the right fuel-air ratio may be visualized by considering the effects of changes in engine speed with a constant throttle setting. If the air flow increases, the venturi suction at the main discharge nozzle increases as the square of the flow and hence the pressure drop across the main jet and the corresponding fuel flow increase in the same manner, maintaining a constant fuel-air ratio. Thus at any given throttle setting the unit behaves in the same manner as a fixed-jet carburetor. By properly proportioning the linkage and contouring the needle the desired fuel-air ratio can be obtained at any and all throttle settings.

The idle system, the choke, and the accelerating pump operate in much the same manner as their counterparts in the fixed-jet carburetor discussed previously. An economizer is not required, for the metering pin can be contoured to give richer mixtures at and near full throttle.

The principal disadvantage of this type of carburetor is its sensitivity to wear, the parts susceptible being the metering pin where it slides in the jet, the jet itself, and the controlling linkage. These factors tend to give mixtures that become progressively richer with use.

SPECIAL METERING SYSTEMS FOR SPARK-IGNITION ENGINES

While the greater part of the spark-ignition engines in use employ float-type carburetors essentially similar to one of the two described above, other types of fuel-metering system are used. In aircraft, for example, the wide range of aircraft attitudes (including inverted flight) in which the carburetor should meter properly make a diaphragm much superior to a float as a means of regulating fuel pressure. Another feature included in the carburetors for most of the larger

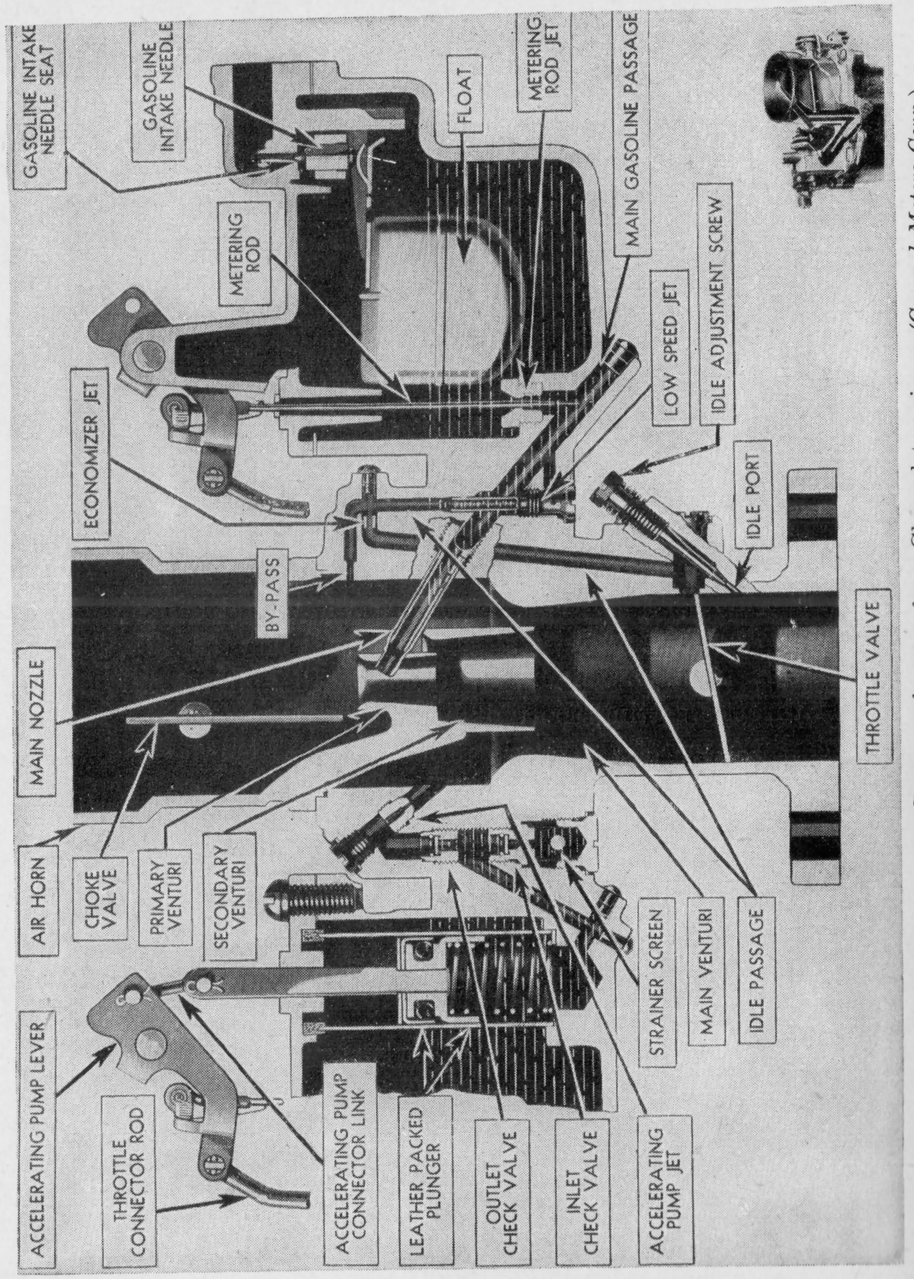


FIG. 101. Section through a Carter carburetor for a Chevrolet engine. (*General Motors Corp.*)

aircraft engines is a device to prevent the carburetor from "richening up" at altitude. Such a device compensates for the increase in venturi suction resulting from a decrease in air density for a given air-weight flow rate—a problem indicated by Eq. (10).

Fuel-injection systems are used in place of carburetors in some models of both large and small aircraft engines. There are two main types of these fuel-injection systems. In one type, the fuel is injected directly into the combustion chamber. In the other, the injection nozzle is located in the intake port, and the fuel is injected into the air stream as it flows through the port into the cylinder. The latter has the advantage that lower injection pressures can be used and the nozzle is not subjected to the high pressures and temperatures prevailing in the combustion chamber. The former arrangement has the advantage that it gives better spray dispersion and prevents liquid fuel from adhering to the intake port walls, from which it may pass into the cylinder somewhat irregularly. Both types of fuel-injection system give good mixture distribution to the various cylinders irrespective of intake manifold temperature and thus give higher volumetric efficiencies under low air-temperature conditions.

As pointed out at the beginning of this chapter, the great majority of spark-ignition engine fuel-metering systems meter on the basis of air flow rate and depend on a venturi-type metering element. It has long been recognized, however, that the engine itself constitutes a meter in that it acts as a constant-displacement pump. At any given rpm the *volume* of air induced each cycle remains substantially constant so that the *weight* of air becomes directly proportional to the intake manifold pressure. In aircraft engines, the wide variations in air density at the carburetor under various altitude conditions make it very difficult to obtain good metering characteristics from a carburetor. For such an application a metering device depending on engine rpm, intake manifold pressure, and intake manifold temperature has many advantages.[3] While the problems of such a device are too specialized to be presented here, the material given in Chap. XIII on the air-pumping characteristics of an engine covers the characteristics of the engine as an air meter.

FUEL-INJECTION EQUIPMENT FOR DIESEL ENGINES

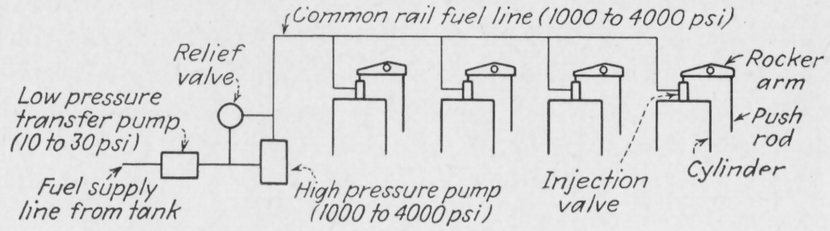
Engine Requirements. From the standpoint of basic requirements, the relations between fuel flow and engine output are quite simple for diesel engines in contrast to the complexity of the corresponding relations for spark-ignition engines. The amount of air

charge taken into a diesel cylinder each cycle is almost independent of either speed or load over the entire operating range. Thus the indicated output of each cycle is approximately proportional to fuel flow rate (see Fig. 81), and hence the power output can be controlled simply by controlling the fuel flow without regard to the air flow rate. From the standpoint of the detail requirements, however, diesel engines present problems that can be overcome only with expensive equipment. In the first place, the fuel must be injected into the cylinder against the high pressures prevailing toward the end of the compression stroke and during the first portion of the combustion process. In the second place, that fuel must be injected in such a way that it will be distributed uniformly throughout the combustion chamber in a very finely divided form. Finally, the amount of fuel injected into each cylinder each cycle must be controlled very accurately so that there will be negligible variations in power output from cycle to cycle and from cylinder to cylinder. This is absolutely essential to smooth engine operation. These requirements make it necessary to employ high injection pressures (upward of 1,000 psi) and injection-pump parts made to accuracies of hundred-thousandths of an inch. Serious problems not indicated above are concerned with such matters as the great sensitivity of combustion rate and character to fuel-spray form and characteristics, injection-nozzle dribbling, and injection timing and duration.

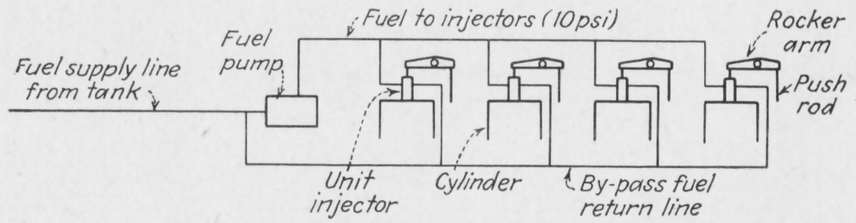
Types of Injection System. Diesel fuel-injection devices can be classified into several groups and subgroups. All types can be classified as either *air* or *solid* injection equipment. In the former, air under a pressure of 2,000 to 4,000 psi is employed to blow fuel into the combustion chamber. In the latter, the fuel remains unmixed with air and free of bubbles until it leaves the injection nozzle—hence the term "solid injection." In both types the nozzle breaks the fuel stream up into a very finely divided spray. Air injection systems have been passing out of use and are not employed in any automotive equipment, largely because of the extra complication, expense, and power requirements of the air compressor.

Solid injection fuel-metering systems may be of the *common-rail* type, in which the fuel-supply pump maintains a high pressure in a line common to all cylinders, as in Fig. 102*a*, or they may be of the type in which each engine cylinder has its own individual plunger-type pump, as in Fig. 102*b*. In the common-rail systems the injection nozzles are opened mechanically, usually by a push-rod–rocker-arm mechanism operated by the camshaft. In the individual-pump

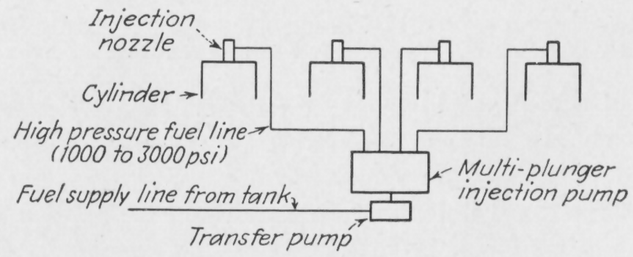
type of system, a spring-loaded valve prevents injection until the pump has built up the proper fuel pressure. In such systems, the pump plungers may be located in a group within a single housing and a supply line run from each element to the cylinder that it supplies,



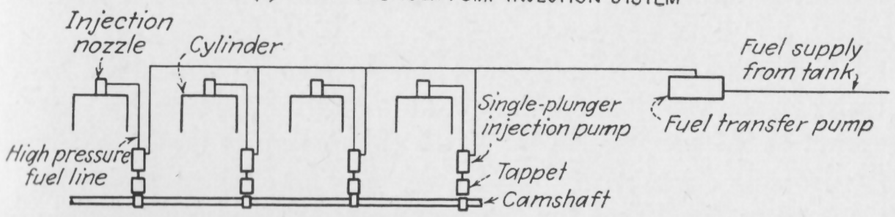
(a) COMMON RAIL FUEL INJECTION SYSTEM



(b) UNIT INJECTOR FUEL INJECTION SYSTEM



(c) MULTI-PLUNGER PUMP INJECTION SYSTEM



(d) INJECTION SYSTEM WITH INDIVIDUAL PUMPS AND SEPARATE NOZZLES

FIG. 102. Types of diesel fuel-injection systems.

as in Fig. 102*c*. This concentrates the control of the amount of fuel delivered by each plunger at one point. If individual pumps in separate housings are used, they may be mounted on the cylinder head and incorporated in a single unit with the nozzle, as in Fig. 102*b*, or

they may be mounted close to the camshaft, as in Fig. 102*d*, and connected to the injection nozzle by lines of moderate length.

Types of Injection Nozzle. A number of types of injection nozzle are in common use. The simplest form of nozzle consists of no more than a small orifice through which the fuel is injected into the cylinder. An elaboration of this form is the multiple-orifice nozzle. Examples of these two types are shown in Fig. 103. In contrast with these nozzles, in which the flow passage area remains fixed, there is another group of *pintle* or *variable-orifice* nozzles such as that in Fig.

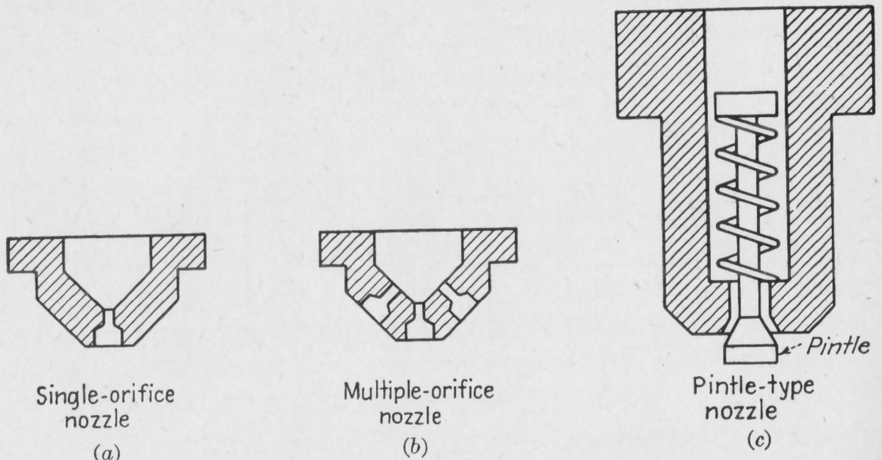


FIG. 103. Types of fuel-injection nozzles.

103*c*. Each of these types has its own peculiar advantages and disadvantages, which can be best understood after an examination of the problems attending fuel-spray formation.

Factors Affecting Fuel-spray Formation. The ultimate goal in diesel fuel-injection work is to distribute a very finely divided fuel spray throughout the air charge in the clearance volume during the short time available for that purpose. This means that both fuel-droplet dispersion in the spray and the distance to which the spray penetrates in the combustion chamber are of vital importance. Some conception of the mechanism by which the stream of liquid is broken up into droplets is essential to an understanding of these problems.

It has been found that the two principal factors acting to break a free stream of liquid fuel down into droplets are turbulence in the stream and fluid frictional forces between the liquid fuel and the air into which it is injected.[4–6] The effects of air density on spray dispersion and penetration are shown in Fig. 104, which forcibly demonstrates the importance of air frictional forces. Figure 105 shows the effect of nozzle shape on spray formation. The three nozzles were

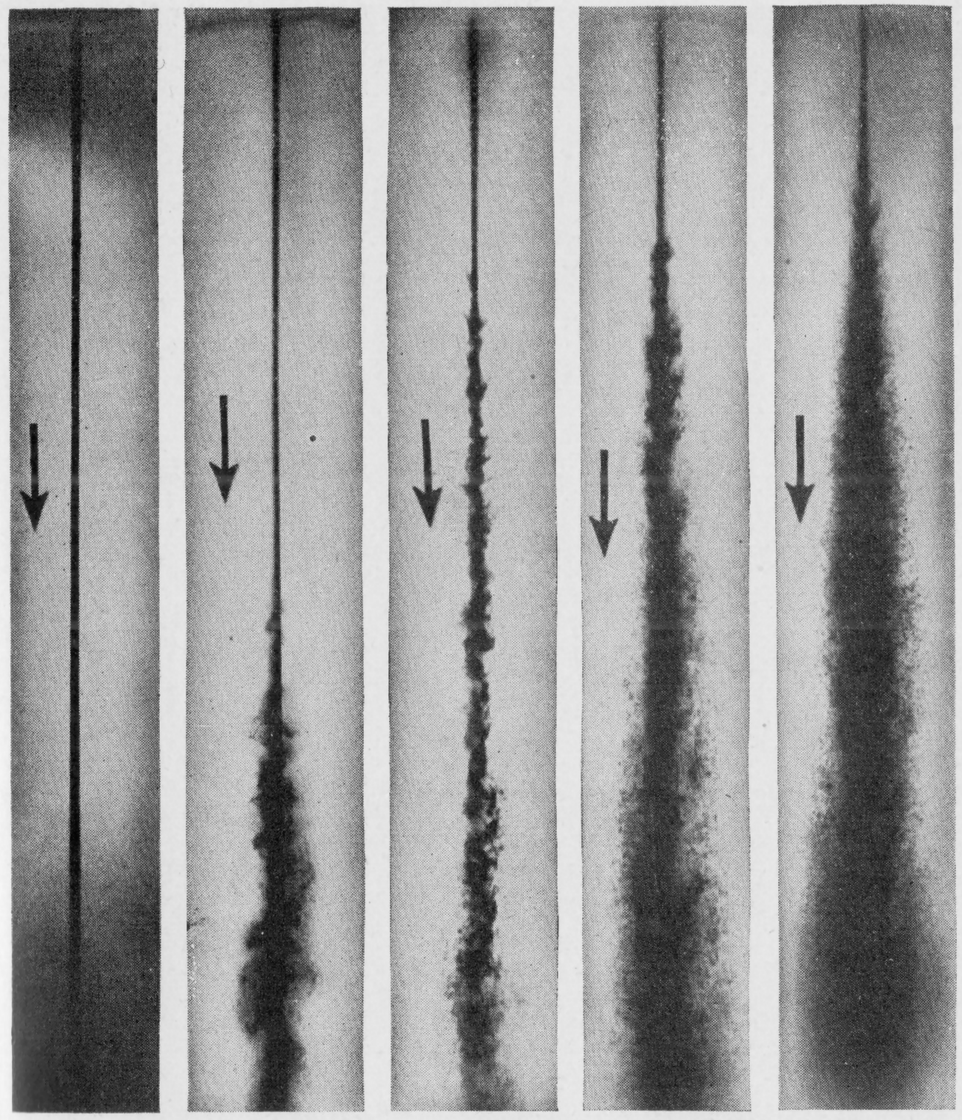
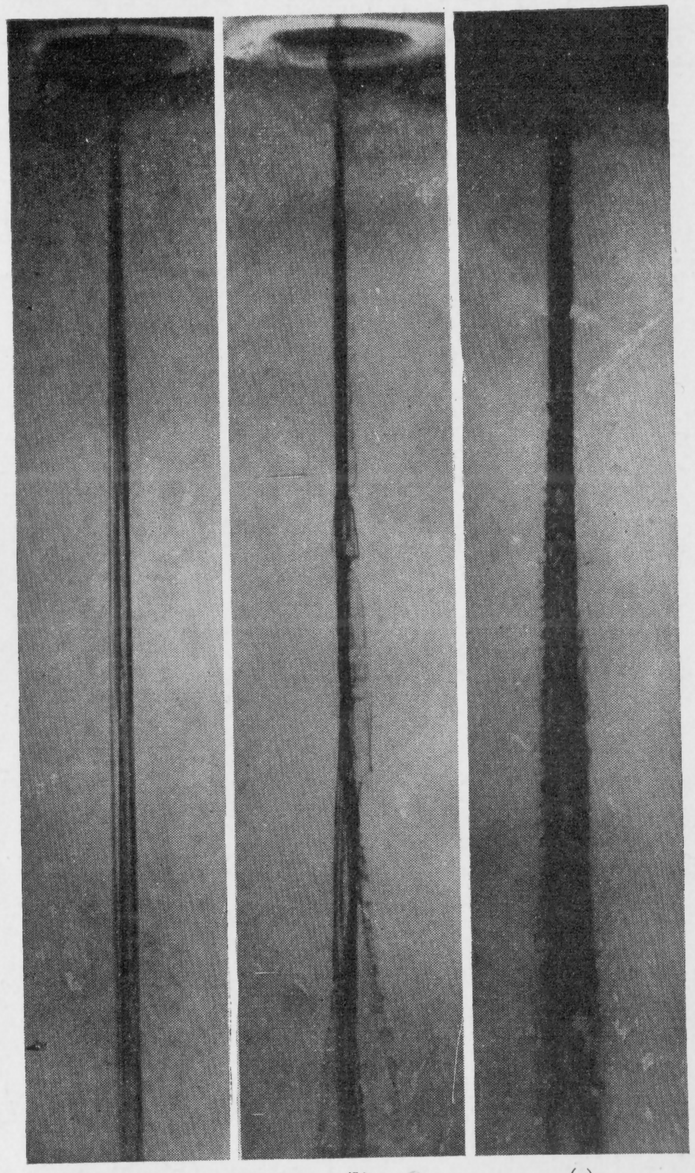


FIG. 104. Effect of air density on fuel-spray penetration and dispersion for a single-orifice injection nozzle. Fuel oil injected under an effective pressure of 250 psi through a 0.020-in. orifice. Air densities, (*a*) 0.0013 atmosphere, (*b*) 1 atmosphere, (*c*) 4.4 atmospheres, (*d*) 7.8 atmospheres, (*e*) 14.5 atmospheres. (*Lee and Spencer, NACA Tech. Rept.* 454, 1933.)

similar to that in Fig. 103*a* and differed only in the contours of the orifice proper. The first nozzle had a smoothly rounded approach, while the other two had the annular chamfer of Fig. 103*a*. The first and second nozzles were smoothly polished, whereas the third was tested without disturbing the small burrs and surface irregularities left by machining. It is evident from the pictures that turbulence in the

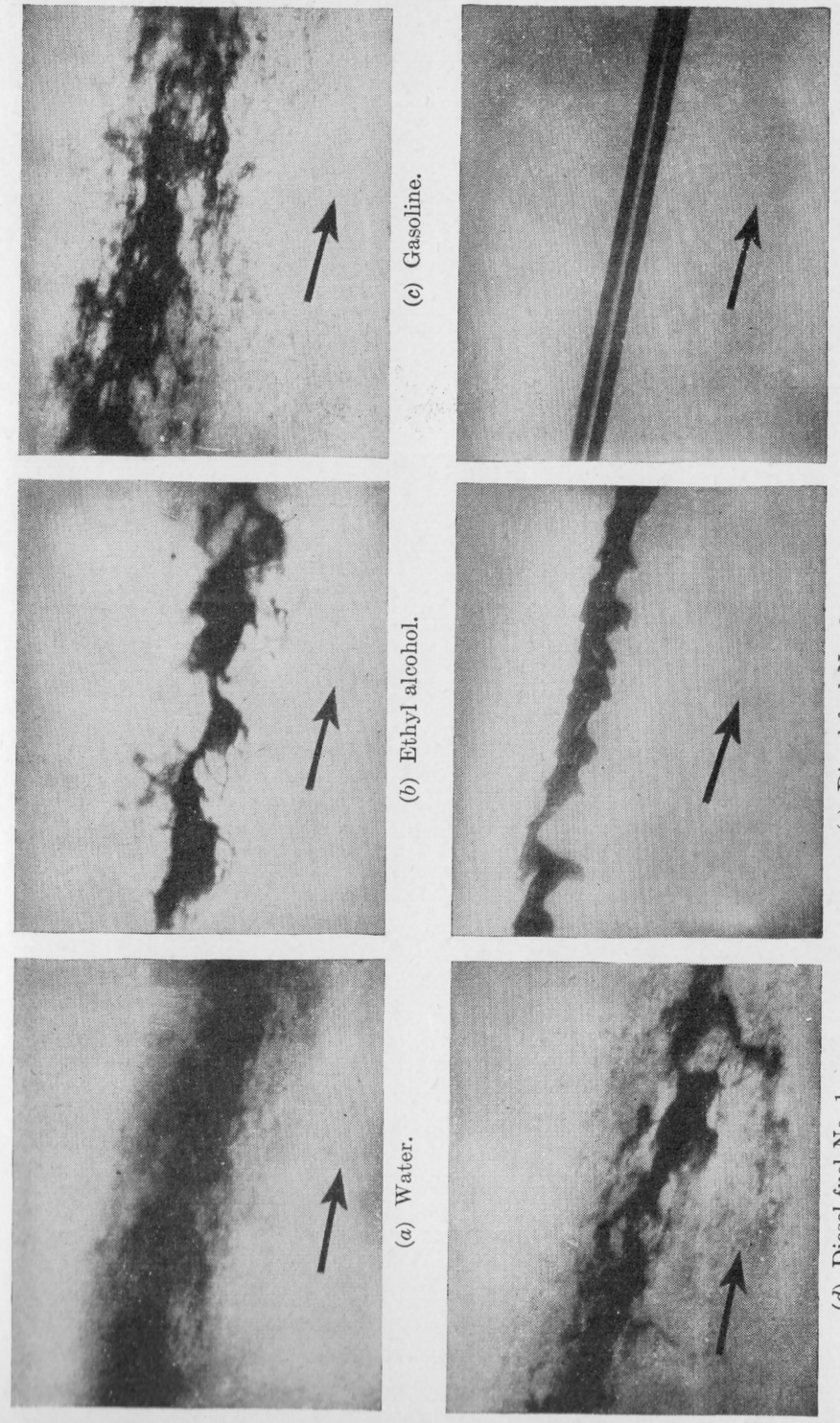


(*a*) (*b*) (*c*)

FIG. 105. Effect of turbulence in the fuel stream on fuel-spray dispersion. Single-orifice nozzle 0.014 in. in diameter and 0.028 in. long; injection pressure, 3,000 psi; air density, 0.0013 atmospheres. Nozzles, (*a*) smoothly rounded approach orifice, (*b*) conventional nozzle, polished, (*c*) conventional nozzle, left unpolished. (*Lee and Spencer, NACA Tech. Rept.* 454, 1933.)

fuel stream assists dispersion but is much less effective than air friction in breaking up the fuel stream.

The viscosity and surface tension of the liquid injected through the nozzle have an important effect on the size of the droplets formed. Figure 106 shows the combined effects of these factors for a number of common liquids.

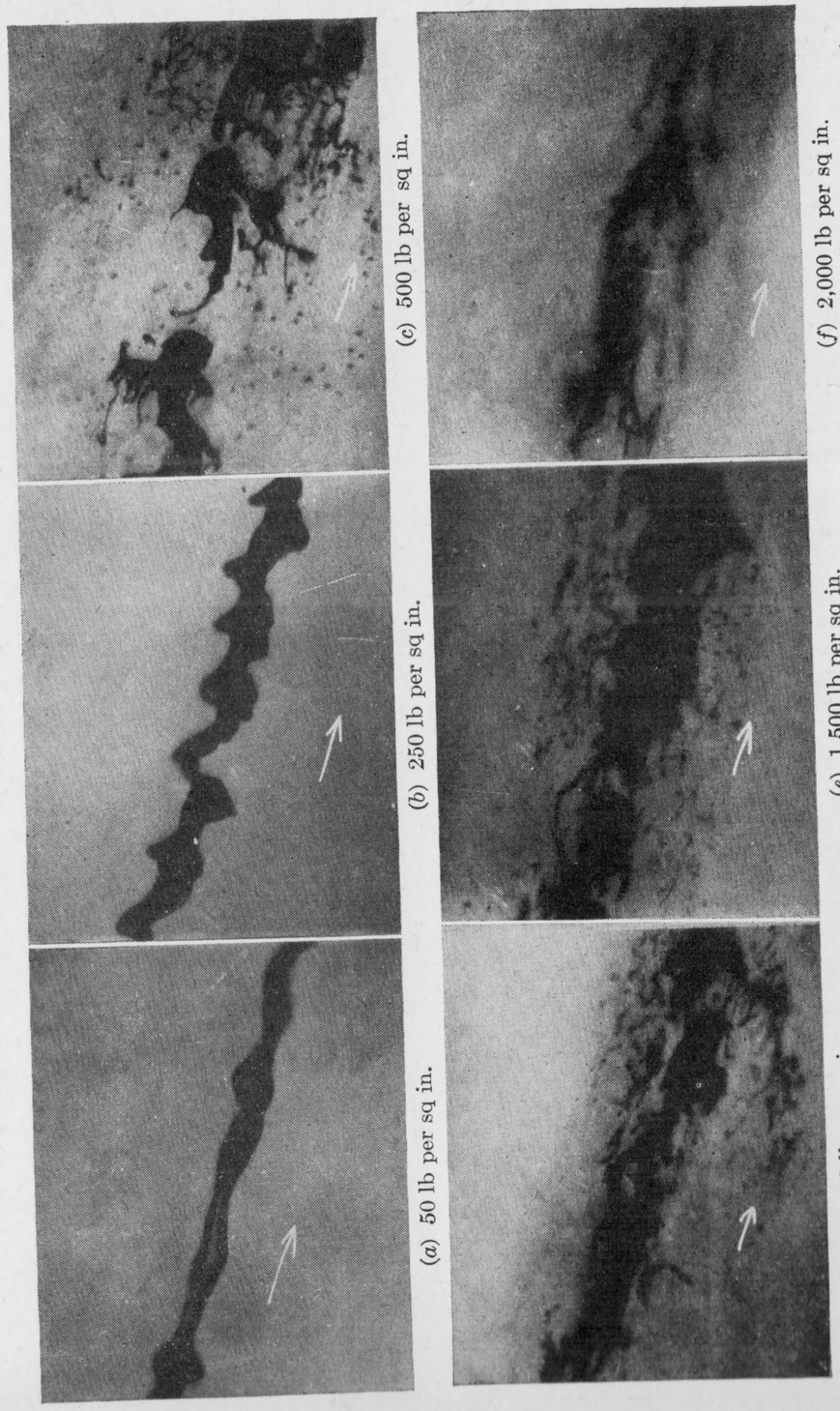


(*c*) Gasoline. (*f*) Lubricating oil.

(*b*) Ethyl alcohol. (*e*) Diesel fuel No. 2.

(*a*) Water. (*d*) Diesel fuel No. 1.

FIG. 106. Effect of fuel viscosity and surface tension on fuel-spray dispersion. Injection pressure, 1,500 psi; orifice diameter 0.020 in.; air density, 1 atmosphere; distance from nozzle, 5 in. (*Lee and Spencer, NACA Tech. Rept.* 454, 1933.)



(a) 50 lb per sq in. (b) 250 lb per sq in. (c) 500 lb per sq in. (d) 1,000 lb per sq in. (e) 1,500 lb per sq in. (f) 2,000 lb per sq in.

Fig. 107. Effect of injection pressure on fuel-spray dispersion. Orifice diameter, 0.020 in.; air density, 1 atmosphere; distance from nozzle, 5.0 in. (*Lee and Spencer, NACA Tech. Rept.* 454, 1933.)

The most important single factor affecting fuel-spray formation that is subject to variation in practice is injection pressure. As might be expected from the increase in the air frictional forces and as shown in Fig. 107, increasing the injection pressure acts to reduce droplet

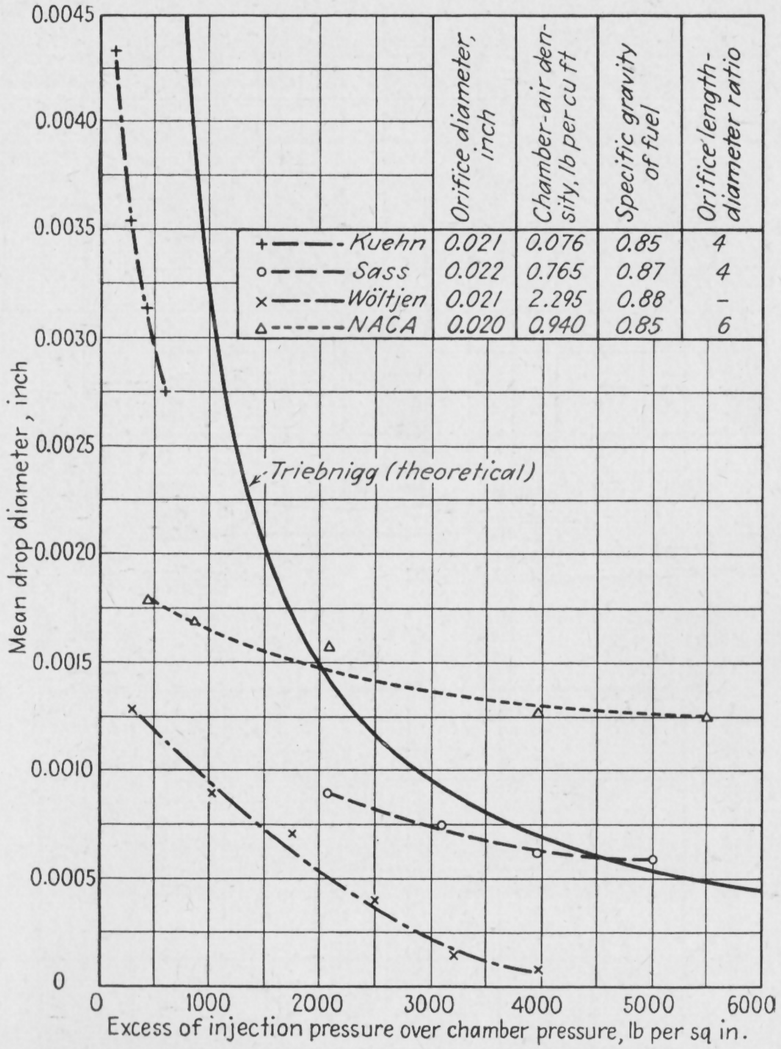


Fig. 108. Effect of injection pressure on mean drop size as determined by a number of different investigators. (*Lee, NACA Tech. Rept.* 425, 1932.)

size and increase spray penetration. Figure 108 shows the average droplet size as a function of injection pressure for typical fuels and nozzles as determined by a number of different investigators. Figure 109 shows the effect of injection pressure on the depth of penetration of the spray into the combustion chamber.

The effect of orifice size on spray formation with a given injection

pressure in the range of interest is rather small so far as droplet size is concerned, as shown by Fig. 110. Increasing the orifice size does tend to increase spray penetration, the magnitude of the increase depending on the orifice size and the injection pressure.

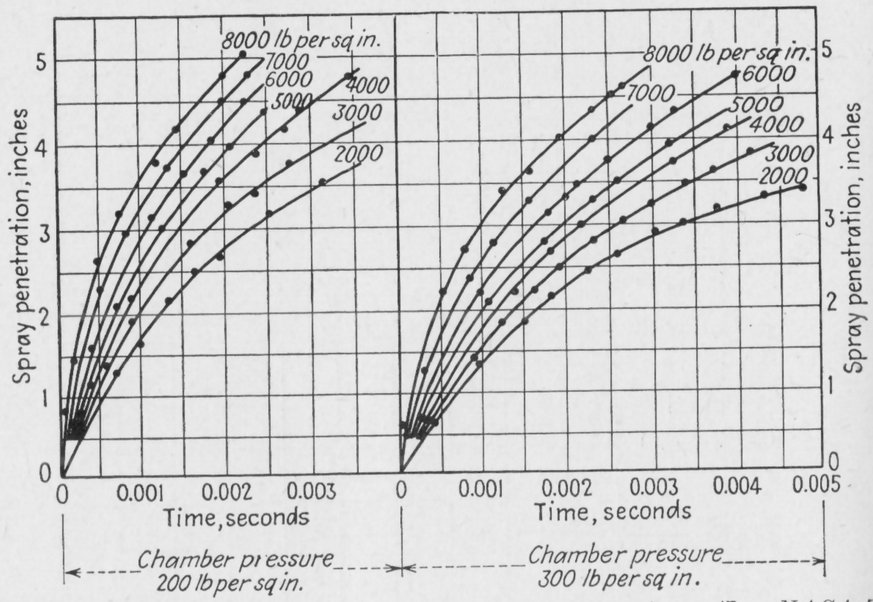


FIG. 109. Effect of injection pressure on fuel-spray penetration. (*Lee, NACA Tech. Rept.* 425, 1932.)

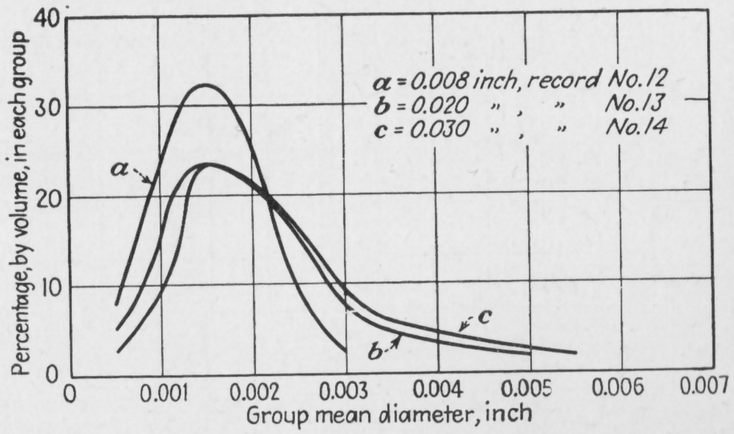


FIG. 110. Effect of orifice diameter on fuel atomization for a mean effective injection pressure of 3,913 psi. (*Lee, NACA Tech. Rept.* 425, 1932.)

The type of nozzle employed also influences the relative effects of operating variables on the form of the spray. Changes in injection pressure will produce considerably greater changes in spray penetration in some types of nozzle than in others, for example. Similarly,

the spray characteristics of one nozzle may be more sensitive to fuel viscosity, air density, etc., than those of another. These effects are very complex and interrelated.[7–9]

Injection-nozzle Characteristics. The type of injection nozzle best suited to any given engine depends on the particular form of combustion chamber employed. In general, a broad, flat spray with low penetration is required by an open-type combustion chamber.

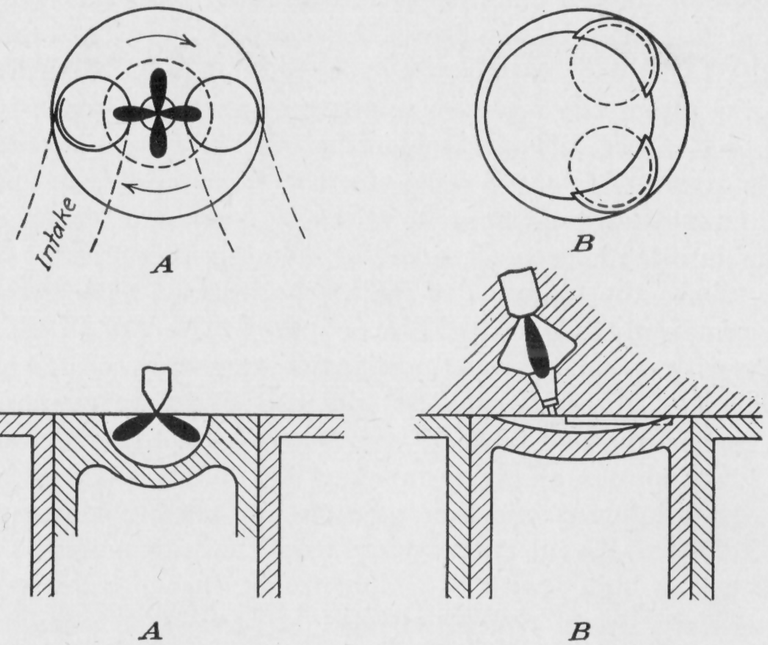


FIG. 111. Types of diesel-engine combustion chambers. (*a*) Open-type combustion chamber; (*b*) precombustion chamber. (*Landen, Trans. SAE, Vol.* 54, 1946.)

Such a spray may be given by a multiple-orifice nozzle for an application such as that in Fig. 111*a*. A precombustion chamber type of engine, however, performs well with a long, slender plume of spray of high penetration such as that given by a pintle-type nozzle for an application such as that in Fig. 111*b*. In all types of combustion chamber, the spray penetration should not be so great that the fuel stream will impinge on the combustion-chamber walls at full load. Not only does fuel that "wets the walls" fail to burn properly, but its incomplete combustion acts to build up objectionable carbon deposits.

The rate at which fuel is delivered from a nozzle is given quite closely by the simple hydraulic relation, viz., the flow rate varies as the square of the pressure differential.[10] If very high pressures are employed, compressibility of the fuel and/or changes in nozzle dimen-

sions may have an appreciable effect. It is also true that viscous rather than turbulent flow may be encountered with heavy fuel oils operating in nozzles in which the orifice length is more than two or three times the diameter. However, the simple hydraulic equation may usually be employed with little error for the lighter fuel oils used in automotive engines. The orifice coefficient of the nozzles is generally high—of the order of 0.90.[10]

Variations in the quantity of fuel injected per cycle with engine load and speed can be handled in various ways. In fixed-orifice nozzles, the fuel quantity injected each cycle may be varied only by increasing either the injection pressure or the duration of injection. This poses rather difficult problems in respect to engines that must operate over a wide speed range, for the duration of injection should be held practically constant in terms of degrees of crank rotation, *i.e.*, the duration in seconds should be inversely proportional to engine speed. Since the quantity of fuel to be injected each cycle at full load is practically independent of speed, the flow rate required increases linearly with speed. Since the injection pressure required varies as the square of the flow rate, an engine designed to operate over a speed range from 400 to 2,000 rpm would have to have twenty-five times the minimum allowable injection pressure at the highest speed. Since the minimum injection pressure for good performance runs about 1,000 psi, the injection system would have to supply as much as 25,000 psi at high speeds if simple fixed-orifice nozzles were used. This difficulty is, of course, avoided in pintle-type nozzles, for an increase in injection pressure brings about an increase in orifice area so that the injection pressure need not rise rapidly with fuel flow rate. Because of the high injection pressures, it has been found in practice that if fixed-orifice nozzles are to be employed the distance between the pump plunger and the nozzle must be kept to an absolute minimum. This is necessary because high pressures cause both an expansion of the fuel-passage walls and a certain amount of compression of the fuel. Inasmuch as the volume of fuel injected each cycle is very small, these effects in even massively designed equipment and tubing seriously interfere with the injection process. The interference takes the form of reduced and erratic injection pressures, excessive injection durations, and nozzle dribbling as the metal walls contract after the injection process has presumably been completed.[11] The *unit injectors* (Fig. 102*b*) in which a pump plunger and a fixed-orifice nozzle are combined in a single unit mounted on the cylinder head have been found to work out very well with fixed-orifice nozzles. Such an

arrangement reduces the space containing high-pressure fuel to a small volume and hence keeps the effects of chamber expansion and fuel compressibility within acceptable limits. Variable-orifice nozzles with the lower injection pressures that they require permit an arrangement such as that of Fig. 102*c*, in which the injection-pump cylinders and pistons are assembled into a single unit for the whole engine with medium-high-pressure lines between the pump and the nozzles.

Pintle-type injection nozzles have a number of important characteristics that should be mentioned. To prevent nozzle dribbling and to ensure good dispersion in the spray, the pintle is spring-loaded so that the pintle valve will not open until a certain minimum pressure is exceeded. This pressure is termed the *pop-off* pressure and may be determined on a bench by gradually increasing the pressure on fuel supplied to the nozzle until small quantities of fuel begin to discharge in little bursts, *i.e.*, until the nozzle begins intermittently to "pop off." Incidentally, if the rate at which fuel is supplied to a pintle nozzle is less than the minimum rate required to keep it well opened up, the pintle valve will chatter on its seat, causing fuel-pressure fluctuations in the nozzle. While these fluctuations tend to give poorer fuel dispersion in the spray, they do make it possible to operate at lower injection rates with a nozzle of a given capacity than would otherwise be the case.

AN INJECTION SYSTEM EMPLOYING UNIT INJECTORS

Applications and Part Arrangement. The unit injector described in the following paragraphs is widely used in diesel trucks, busses, earth-handling equipment, railroad locomotives, ships, and stationary power plants. It is employed in the two-stroke-cycle engine shown on pages 22 and 23. A section through the injector and closely related parts of that engine is shown in Fig. 112. The injector is operated from the camshaft by a push rod and rocker arm. It is mounted in a copper tube that extends through the coolant cavity in the top of the cylinder head, the ends of the tube being swaged into the head to give water- and gastight seals. This arrangement is employed instead of a boss in the cylinder head to provide better cooling of the injector. The injector fuel-delivery rate is controlled by a tubular shaft at the right of Fig. 112. Torsional motion of the shaft is transmitted through a lever and link to the injector, where a rack integral with the link engages and rotates a gear controlling the effective stroke of the pump plunger.

Construction of the Injector. The constructional details of the unit injector itself are shown more clearly in the detail cutaway view of Fig. 113. The heart of the injector is the *plunger* and *bushing*, which are the piston and cylinder of the pump. The plunger is actuated by the rocker arm through the intermediate *follower*. A *spring* returns the plunger and follower to their upper positions

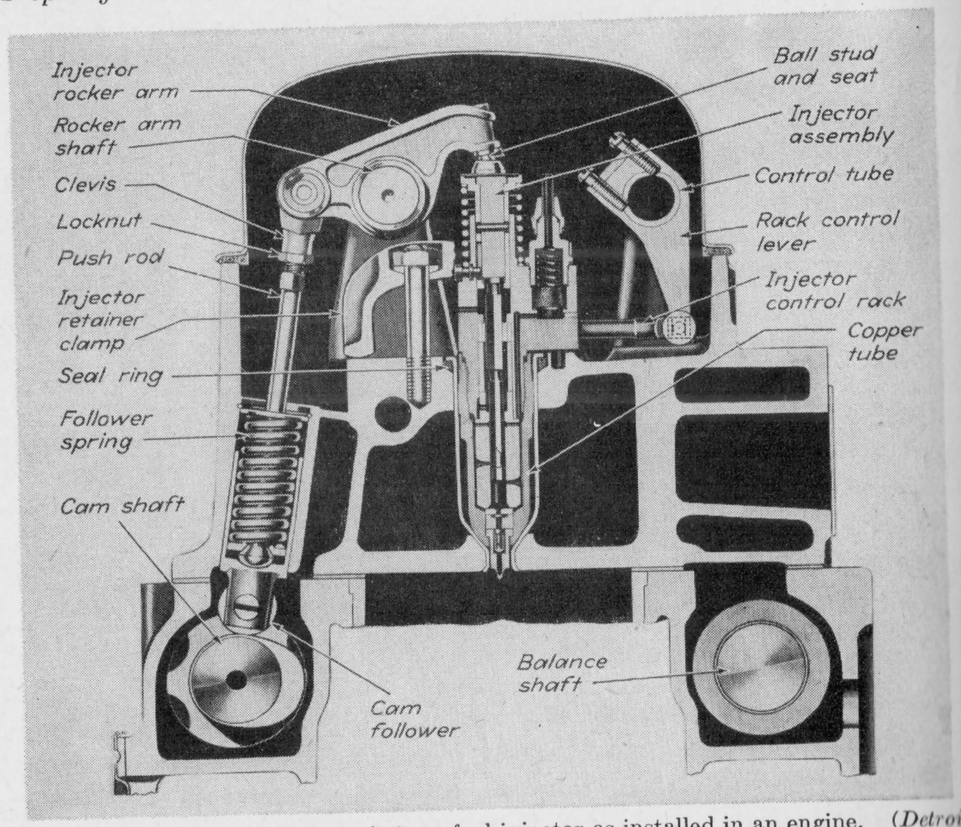


FIG. 112. Section through a unit-type fuel injector as installed in an engine. (*Detroit Diesel Engine Division, General Motors Corp.*)

at the end of injection. The *gear* keyed to the plunger transmits motion from the *rack*, rotating the plunger to change the relationship between the helical groove at its lower end and the upper fuel port in the bushing, thus controlling the amount of fuel injected. (The details of the manner in which this effects control will be discussed in the next paragraph.) As the plunger moves downward, the fuel is forced through a *check valve*. This check valve is designed to open and permit fuel to flow downward; but if combustion-chamber gases tend to flow back upward into the injector during the period following injection, this check valve closes to prevent the backflow. The fuel passes from the check valve to a *spray-tip valve*, which

is designed to prevent dribbling and which will not open until the pressure builds up to a value sufficiently high to ensure good spray characteristics. The valve setting for the injector is usually made to give a pop-off pressure of 600 psi.

Metering of the Fuel. As pointed out above, the fuel is metered by changing the *effective* stroke of the plunger, which operates with

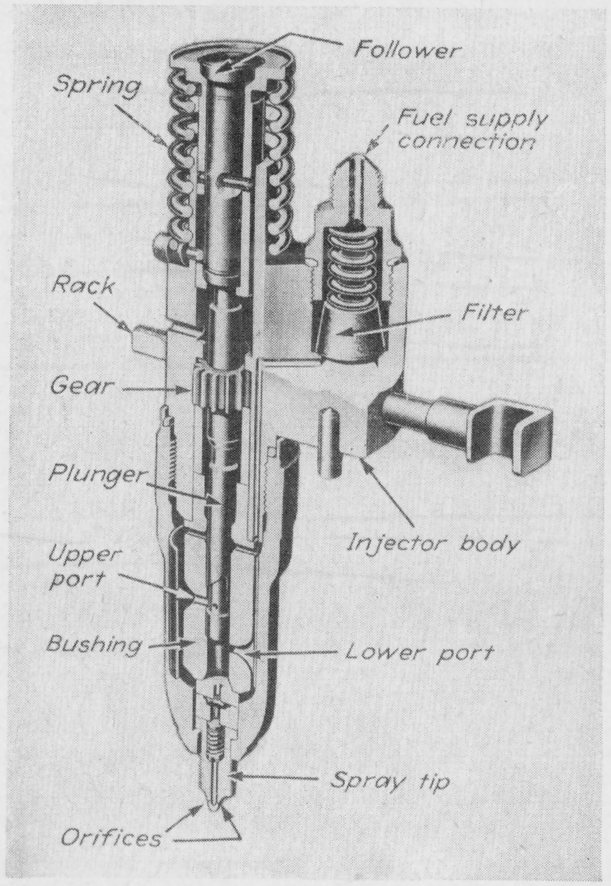


FIG. 113. Section through a unit-type fuel-injector assembly. (*Detroit Diesel Engine Division, General Motors Corp.*)

what is, of course, actually a constant stroke. The effective stroke is reduced by leaving an intake port open until the plunger has traveled a considerable distance on its downward path. Thus no appreciable pressure is built up by the plunger, and hence no fuel is injected until the intake port is closed. The action can be visualized by examining Fig. 114. An annular groove is provided around the plunger just above its lower end. A set of drilled holes provide a passageway for oil to flow from the annular groove down through the center of the plunger to the space below its lower end. The cutaway view at the

left of Fig. 114 shows the plunger at the top of its stroke, with both the upper and lower ports uncovered. (As is evident in Fig. 113, both ports are connected to the fuel-supply line by passages in the body of the unit.) As the plunger is moved downward a short distance, it closes off the lower port. Further downward motion displaces oil up through the annulus and upper port until the upper edge of the annulus drops below the edge of the upper port. The plunger then builds up pressure in the fuel beneath it and forces it out through

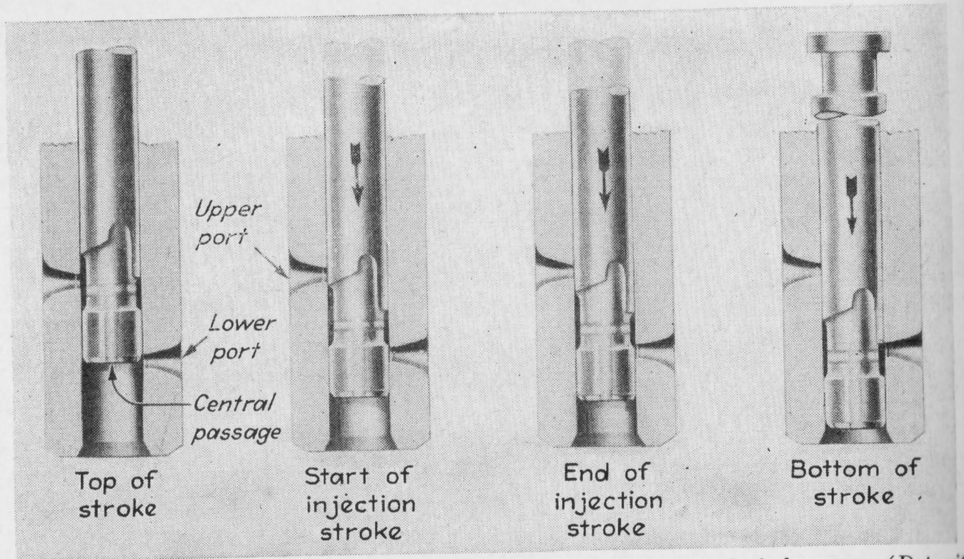


FIG. 114. Phases of unit-type injector operation by vertical travel of plunger. (*Detroit Diesel Engine Division, General Motors Corp.*)

the spray-nozzle tip. The injection process continues until the plunger has moved downward to the point where the annulus uncovers the lower port. Further movement of the plunger to the bottom of its stroke serves only to flush fuel out of the cylinder and send it back through the lower port. As is evident in Figs. 113 and 114, the upper and lower edges of the annulus in the plunger are helical so that rotation of the plunger varies the timing of the beginning and the end of injection. Since the two are varied at different rates, the quantity of fuel metered to the cylinder is also varied.

The injector is supplied with fuel at a pressure of about 20 psi, while a return line is also provided so that a cooling stream of fuel is constantly circulated through the passages in the body of the unit. Filters in both the supply and the return lines trap any dirt particles that might tend to clog the orifices in the spray tip, prevent proper closing of the check or spray-tip valves, cause sticking or scoring of the plunger, or give other similar troubles.

AN INJECTION SYSTEM EMPLOYING A MULTIPLUNGER PUMP AND PINTLE-TYPE NOZZLES

Applications and Pump Construction. The injection-system components described in the following paragraphs are used on a variety of truck, agricultural, and marine diesel engines. Somewhat similar units have also been employed for fuel-injection spark-ignition engines for both aircraft and truck use.

Figure 115 shows a picture of a complete pump unit. Note that a governor and an automatic-injection advance mechanism have been installed in a housing attached to the top of the basic metering and injection pump, while a fuel-transfer pump has been attached in a similar fashion to the bottom. The complete unit is attached by means of the flange at the right to a corresponding mounting pad on the engine. Engine control is obtained through the lever at the upper right, the action being transmitted through the governor so that the lever shown is primarily a speed rather than a power control. Fittings for the tubes that carry fuel to the individual nozzles can be seen at the left end of the pump, where they are placed radially about the drive-shaft axis with a uniform spacing. These fittings are at the outer ends of the pump plungers, which are placed with their axes parallel to that of the drive shaft. A swash plate on the drive shaft at the right end actuates the plungers through tappets, a spring on each plunger serving to keep it riding against its tappet and the tappet riding on the swash plate. A rotary distributor valve is located at the left end of the pump concentric with the drive shaft. It is moved axially to vary the fuel delivery rate, the rubber bellows seal evident in Fig. 115 permitting that motion. The nut just outside the rubber bellows permits easy adjustment of the idle setting of this valve.

Fuel Metering. The mechanism of fuel metering is shown in the detail views of Fig. 116. As a plunger moves backward after an injection stroke, fuel is drawn into the cylinder until the plunger has reached its extreme right-hand position, as shown in Fig. 116*a*. Note that the left end of the bore of the cylinder in which the plunger moves is enlarged so that there is always a passage for fuel to flow back from the extreme left end of the cylinder to the inlet port at the bottom, regardless of plunger position. A spring-loaded valve in the outlet port at the upper left prevents fuel from flowing out to the injection nozzle until considerable pressure is built up in the pump. As the plunger moves to the left on the injection stroke to the position shown in Fig. 116*b*, fuel flows back out through the inlet port to the supply line. This continues until the wedge-shaped land on the rotary dis-

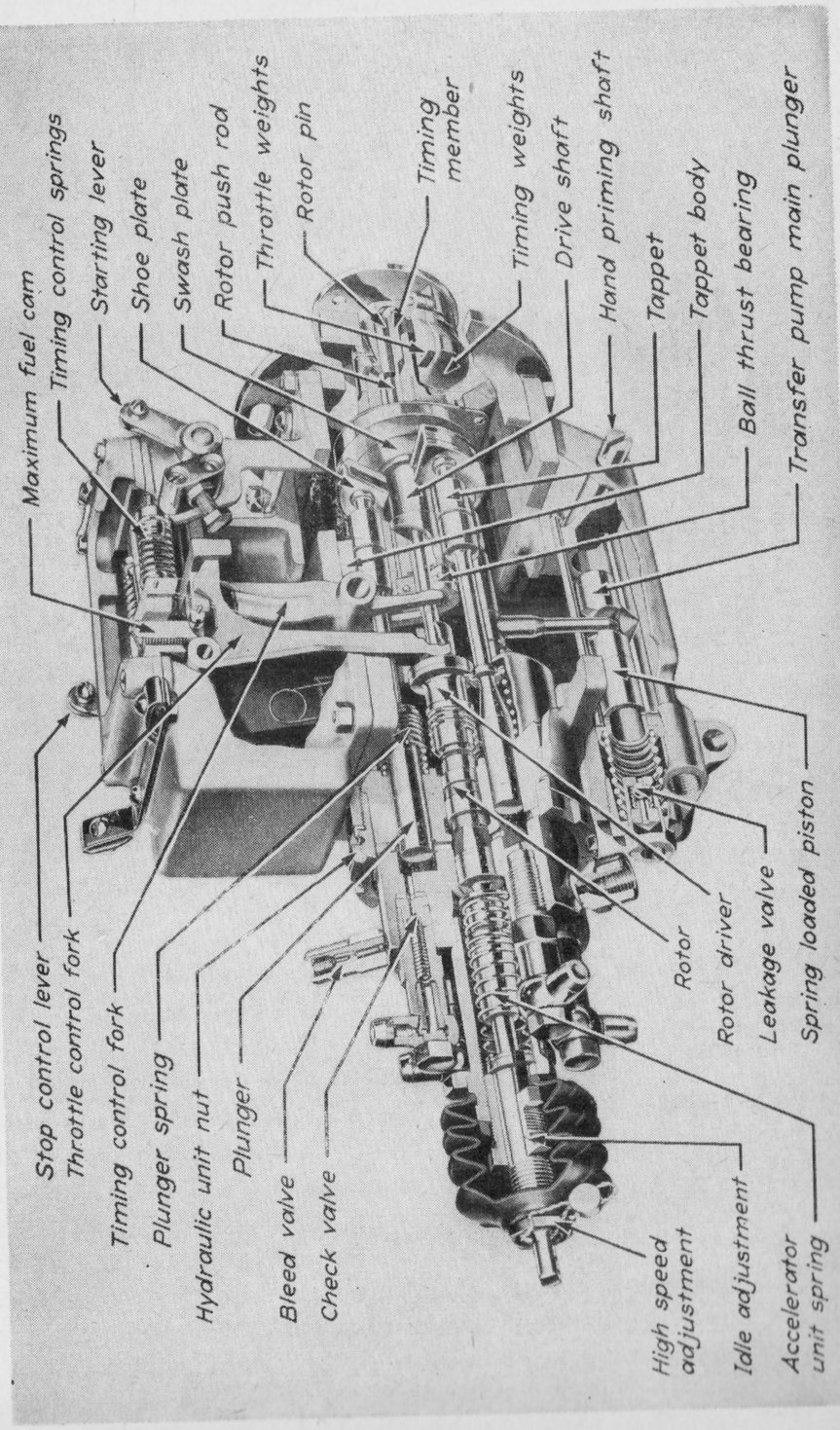


FIG. 115. Multiple-plunger fuel-injector pump for automotive diesel engines. (*Excello Corp.*)

tributor valve closes off the inlet port. Further motion of the plunger then compresses and delivers fuel to the nozzle until the distributor valve has rotated to the point where it again uncovers the inlet port and relieves the pressure under the plunger (see Figs. 116*c* and 116*d*). The length of plunger stroke effective in delivering fuel to the nozzle

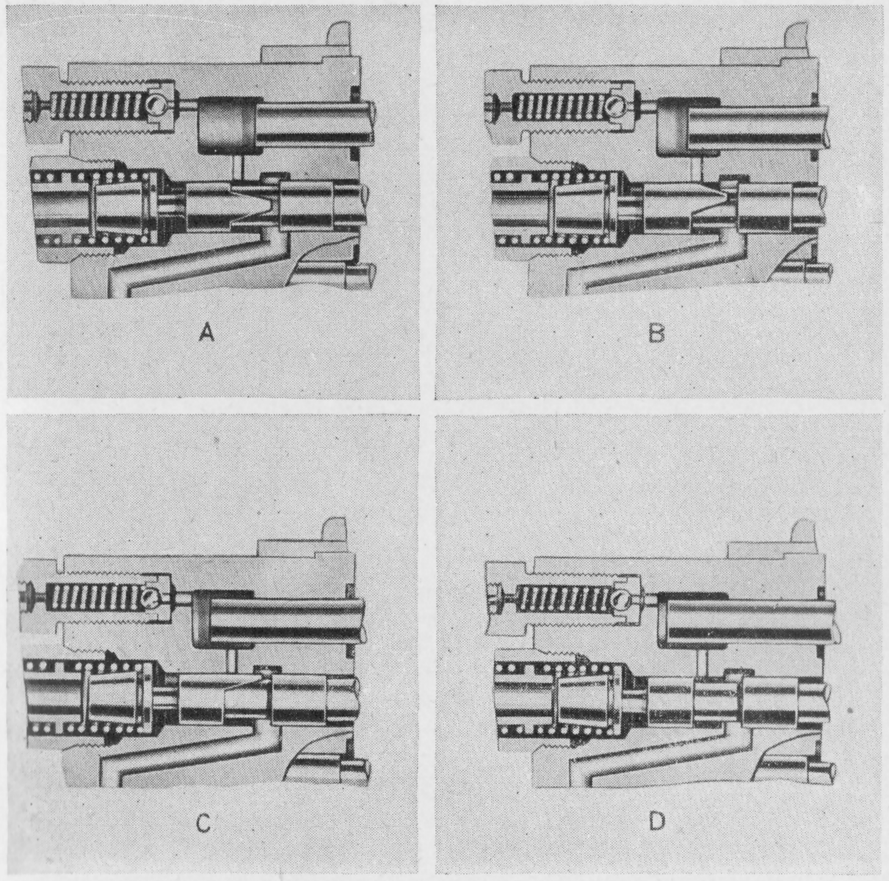


FIG. 116. Phases in a metering cycle for one plunger of the fuel-injector pump of Fig. 115. The cylindrical member in the upper right of each view is the *plunger*, while that beneath it is the *rotary distributor valve*. (*Excello Corp.*)

is controlled by the axial position of the distributor valve. If the valve is pushed to the left, the width of the wedge-shaped land wiping the ports as it rotates is relatively large and the amount of injection correspondingly great. Similarly, sliding the distributor valve to the right reduces the width of the land closing off the port and hence reduces the fuel flow rate.

Construction and Operation of the Pintle Nozzle. The nozzle employed with the above injection pump is shown in section in Fig.

117. The circular insert shows the details of the spray-tip assembly. A bracket-type clamp is used to secure the nozzle against the engine, a gastight seal being provided by a copper gasket at the cylinder-head end of the nozzle.

Fuel flowing into the nozzle body passes unimpeded into the tip assembly to a filter screen. While this is not definitely shown in Fig. 117, holes are provided in the spring hanger so that the fuel can also

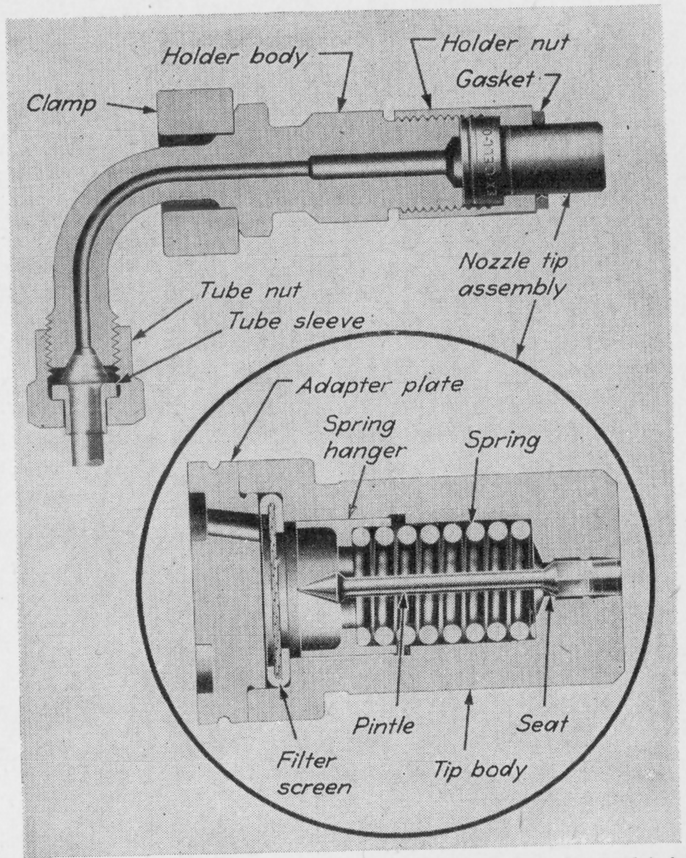


FIG. 117. Details of construction of the injection nozzle for the fuel-injector pump of Fig. 115. (*Excello Corp.*)

pass freely from the screen to the pintle-valve seat at the right end of the nozzle tip. Again while this is not definitely shown, the spring and spring hanger act on the left end of the pintle to exert considerable force to keep the valve seated. When the fuel-injection pump sends fuel to the nozzle under a pressure of about 2,000 psi, however, the pressure forces the pintle to the right off its seat and permits fuel to flow through the annular orifice formed at the nozzle tip. Since the outer end of the pintle is tapered, the greater the quantity of fuel

delivered by the pump, the greater the pintle-valve lift and the greater the area of the annular orifice.

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Problems

1. If a single-venturi simple float-type carburetor has a 1.125-in.-throat venturi, approximately what size main jet should it have to give a fuel-air ratio of 0.070 in the central portion of its normal operating range of air flow?

2. If a bus carburetor is set to give a fuel-air ratio of 0.070 at sea level and 60°F, what fuel-air ratio should it be expected to give at 10,000 ft on a hot day if under those conditions the barometer runs 20.5 in. Hg and the temperature is 75°F?

3. Sketch a schematic diagram showing the functional relationship between the parts of a float-type carburetor having both a main metering system and an idle system. Label the following parts: venturi, main jet, main air bleed, float, float valve, float bowl, idle air bleed, idle jet, idle needle valve, throttle valve, and throttle stop screw.

4. Determine the amount of fuel in cubic centimeters to be delivered to a 4- by 5-in. diesel cylinder each cycle if the volumetric efficiency is 90 per cent, the inlet air temperature 100°F, the barometric pressure 29.4 in. Hg, and the fuel-air ratio 0.055. Repeat for a fuel-air ratio of 0.012. (These fuel-air ratios would represent full-load and idle conditions, respectively.)

5. Determine the effective stroke of a 0.8-cm-diameter plunger required to deliver the fuel charge each cycle for both conditions of Prob. 4. Neglect leakage effects.

6. Determine the allowable variation in effective plunger stroke between cylinders in Prob. 5 if the fuel flow rate for each cylinder is to be held within ±2 per cent.

7. Determine the increase in volume under injection pressure of the inside of a steel tube carrying fuel from the pump to the nozzle. Assume a steel tube 14 in. long, ¼ in. OD, ⅛ in. ID operating with an injection pressure of 2,000 psi. Compare the resulting value with the answers to Probs. 4 and 6.

8. Estimate the velocity of a fuel jet leaving an injection nozzle if the injection pressure ahead of the nozzle is 2,000 psi and the cylinder pressure is 400 psi. Assume a value of 0.785 for the specific gravity of the fuel.

CHAPTER VII

IGNITION

There are many problems of great practical importance associated with the ignition of the charge in gasoline engines. This chapter presents the more important of these problems from the mechanical rather than the electrical standpoint, for a thorough treatment of the electrical problems is beyond the scope of this book.

The material covered includes a discussion of the effectiveness of electric sparks in the ignition process, functional descriptions of widely used types of ignition system and their components, and a presentation of the more important operating problems and limitations.

EFFECTIVENESS OF IGNITION SPARKS

Effects of Spark Gap and Intensity. A considerable amount of work has been carried out in an effort to determine the extent to which the character and intensity of a spark affect the ignition process. These tests have revealed that there is very little to be gained by providing more than the minimum spark necessary to ensure inflammation.[1] It has been found that, for automotive engines, a spark gap of about 0.025 in. provided with a sparking potential about half again to twice as great as the minimum required to jump that gap gives very good performance over the entire speed range.[2] While a lower sparking potential would be satisfactory much of the time, some difficulty with misfiring would be encountered occasionally under adverse combinations of circumstances, as will be evident from the following discussion.

Factors Affecting Sparking Potential. If a spark gap and a set of equipment are once selected so that the electrical characteristics of the ignition system are determined, it has been found that there are many factors which influence the voltage required at the spark plug before a spark will jump across the spark-plug gap. These include electrode temperature, engine rpm, engine compression ratio, intake-manifold pressure, fuel-air ratio, and amount and nature of lead-compound deposits on the electrodes. All these items influence the ionization of the gases between the electrodes and hence the voltage

required for a spark. The effects of both spark gap and engine rpm on the sparking potential at the spark plug are shown in Fig. 118 for a representative engine. Note that the effect of the length of the spark gap is much less than might be expected, for the sparking potential increases relatively little instead of in direct proportion to the gap length. Similarly, while an increase in gas density in the spark gap should increase the required sparking potential, this is not always the case, for other factors such as electrode temperature and the character of the deposits on the electrodes cannot be maintained constant. As a result, while the sparking potential required usually increases with

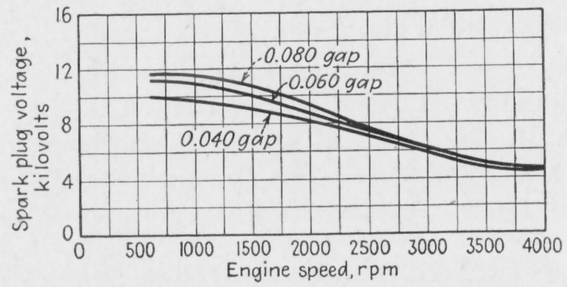


Fig. 118. Sparking potential as a function of rpm for several spark gaps in an automotive engine. (*Hartzell, Trans. SAE., Vol.* 53, 1945.)

either engine load or compression ratio, this does not always happen, for the electrode temperature, for instance, may also increase at the same time and thus influence the degree of ionization. In fact, there have been numerous instances in which the observed effects of changing a variable such as the manifold pressure have been just the opposite to those expected.

Service experience has demonstrated that the cumulative effects of all operating variables are such that the ignition system should be capable of supplying a sparking potential approximately equal to that indicated by the upper curve in Fig. 118. This will ensure freedom from misfiring throughout the engine operating range.

Effect of Heat of Spark. The amount of heat energy released in the spark is related to its effectiveness in igniting the charge. This heat energy is most conveniently measured electrically by determining the voltage drop across the gap and the current flow. Since both factors vary rapidly with time, it is necessary to take oscillographic records and integrate the instantaneous products to obtain the total energy input per discharge. The problem is somewhat complicated by the fact that the spark is usually in the form of a series of oscillating discharges. This effect may be deduced from Fig. 119, which shows curves for both primary current and secondary voltage as given

by a conventional battery-coil ignition system. Depending on the length of the spark gap, the capacitance of the ignition wiring, etc., the spark will oscillate back and forth across the gap until the voltage available is less than that required, at which point the spark will die out.

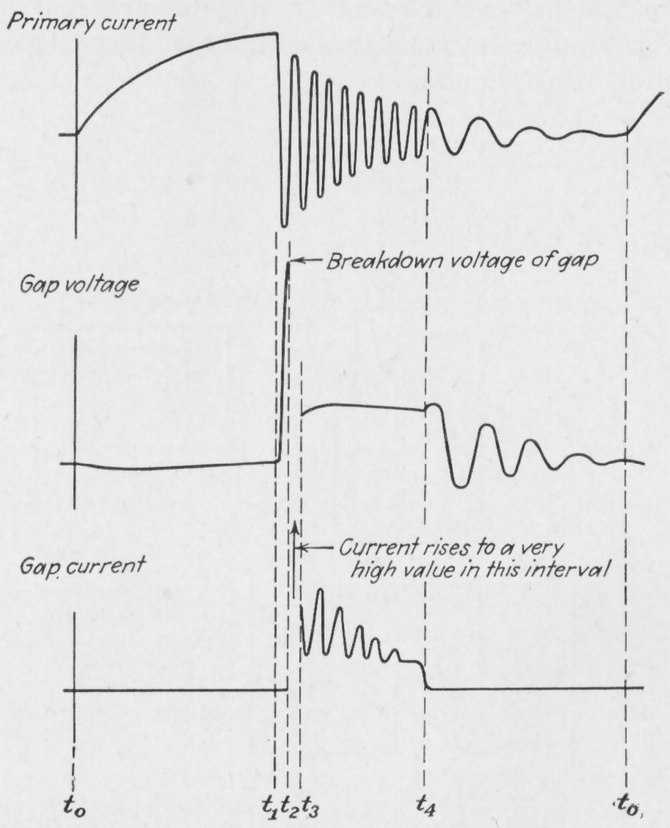


Fig. 119. Character of the relations between primary current, gap voltage, gap current, and time for a battery-coil ignition system. (*Plesset, McCrumm, Plesset, and Floyd, Mech. Eng., Vol.* 67, 1945.)

The total heat energy liberated in the spark gap per ignition cycle need not be great for normal operation—3 to 5 millijoules being sufficient for ignition. When the engine is cold, however, as under starting conditions, a greater amount of energy is needed for an effective ignition spark. Battery ignition systems give 15 to 18 millijoules per spark under starting conditions—an entirely adequate amount. The long oscillating character of the discharge is a major factor in raising the total energy release per ignition cycle to this value.

IGNITION SYSTEMS

The Battery-coil Ignition System. By far the most widely used ignition system is the battery-coil type having the functional layout

indicated in Fig. 120. Most motor-vehicle engines, some of the smaller aircraft engines, and many stationary and marine engines employ this type of ignition system. A battery serves as the electric power supply, while an induction coil is used to give the high voltages required to jump the spark gaps in the engine cylinders. Battery-coil ignition systems have proved to be very reliable and are inexpensive if a battery and generator are required for other purposes such as starting and lighting circuits.

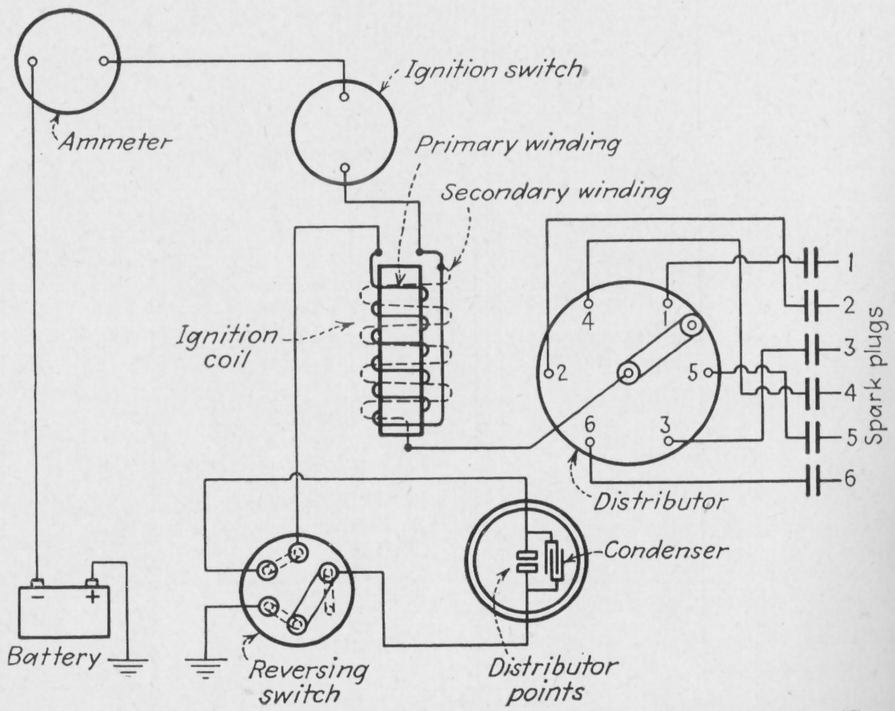


Fig. 120. Schematic wiring diagram for a battery-coil ignition system. (*General Motors Corp.*)

In examining the details of Fig. 120 it should be noted that one of the most critical elements in the system is the set of breaker points which control the flow of current from the battery through the primary winding of the coil. When these points close, a current begins to flow in the primary circuit. This current flow takes an appreciable time to build up, for it is retarded by the counter emf (electromotive force) set up by its own magnetic field. The soft iron core of the coil greatly intensifies this effect, for it increases the strength of the coil's magnetic field. If the breaker points are opened suddenly after the current flow in the primary has reached its maximum, the condenser across them prevents arcing and the d-c current flow in the primary circuit is abruptly stopped. The very rapid collapse of the magnetic

field that accompanies this induces a high voltage in the secondary winding, as shown in Fig. 119. The secondary current flows from the coil through the rotating distributor finger to a spark plug and across the spark gap. Both the breaker points and the distributor finger are carefully synchronized with the crankshaft to give the proper timing of the spark. The inductance and capacitance in the coupled system are responsible for the oscillations in the two circuits after the breaker points open.

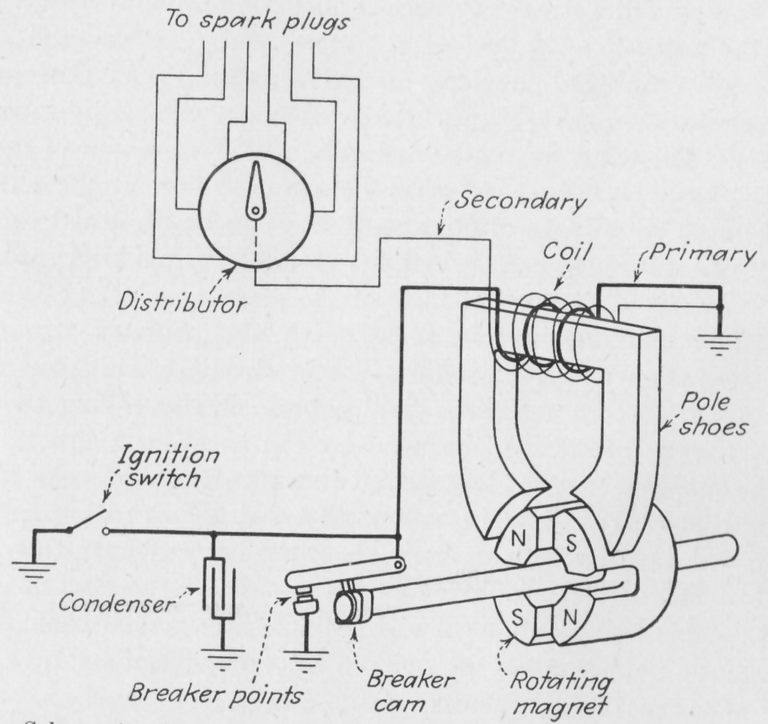


Fig. 121. Schematic wiring diagram for a magneto ignition system. (*General Motors Corp.*)

The Magneto Ignition System. Many heavy-duty and/or high-performance engines employ magneto ignition systems. These engine types include a large number of tractor and stationary engines as well as most aircraft engines. Magneto ignition systems have the advantage that they are not dependent on a battery. This is important if a battery is not required for starting, lights, etc., or where the greatest possible reliability is desired. The magneto ignition system also has the important advantage that the amount of electric power available increases with speed so that there is no tendency for the spark intensity to fall off at high speeds as there is in battery-coil systems.

The essential elements of a magneto ignition system are shown in

Fig. 121. Modern magnetos employ rotating magnets made of an alloy such as alnico that can be made to give an exceptionally strong permanent magnet. As indicated in Fig. 121, such a magnet is rotated between two pole shoes that are connected by an iron core on which both the primary and the secondary coils are wound. A set of breaker points in the primary circuit is opened and closed by a cam on the same shaft as the rotating magnet. As the rotating magnet turns, the direction of the magnetic flux through the soft iron core of the coil reverses direction—a reversal occurring twice for each revolution of the rotor if a two-pole magnet is used. (Four- and six-pole magnets are employed in some aircraft engines.) As the magnetic flux alternately builds up and breaks down, a voltage is induced in both the primary and the secondary coils. The resistance of the spark gap is too great, however, to permit current to flow in the secondary unless a more rapid rate of change of flux can be obtained than that induced by rotation of the permanent magnet. This high rate of flux change is achieved through the use of the primary winding in the coil. When the breaker points in series with the primary winding are closed, they allow a current to flow in that circuit. This primary current sets up its own magnetic field, which acts to retard the breakdown of the magnetic field induced by the rotating magnet. Then, when the rotating magnet has moved to a position such that the magnetic field which it induced is about to break down rather rapidly in spite of the magnetic field from the primary current, the breaker points are timed to open abruptly and terminate the flow of primary current. This brings about a virtually instantaneous breakdown of the magnetic field around the coil, so that an extremely high rate of change of magnetic flux results. This, of course, induces a very high voltage in the secondary winding of the coil. The induced voltage runs from 10,000 to 15,000 volts, more than ample to cause a good spark to jump across the gap between the spark-plug electrodes and ignite the charge in the cylinder.

A number of features of magneto ignition systems should be noted in examining Fig. 121. In the first place, the ignition switch serves to short-circuit the breaker points when in the "off" position, thus preventing a rapid breakdown in magnetic flux. Unlike the ignition switch in the battery-coil ignition system, it does not stop the flow of current in the primary circuit. In the second place, the rotating magnet must be geared to the engine with a ratio such that the number of sparks required for the engine each revolution will be equal to the number of flux reversals given by the rotating magnet in the same interval of time. Thus a four-cylinder four-cycle engine employing a magneto with a two-pole magnet would have a 1:1 ratio between the crankshaft and the magneto rotor. Similarly, the same magneto would have to run at twice engine speed if it were to be used on an eight-cylinder engine.

As implied above by the statements regarding the importance of the rate of change in magnetic flux, a magneto will not fire the spark plugs until a certain minimum speed known as the *coming-in speed* is reached. In the basic type of magneto of Fig. 121, this speed usually corresponds to about 100 engine rpm. This means that a relatively high cranking speed is required to start the engine. The coming-in speed can be reduced and the spark retarded to prevent "kickback" of the engine during cranking by employing an *impulse coupling.* These units are constructed so that at low speeds the magnetic rotor is momentarily stopped while the rotation of the drive shaft builds up tension in a spring between the two. At the proper instant the rotor is released, so that, driven by the taut spring, it rotates several times its average speed to give a higher rate of change of flux. A simple flyball-governor type of mechanism is incorporated in the impulse coupling so that it does not operate at speeds above the cranking range. The mechanism is usually made to give a retarded spark when the impulse coupling is operative. Magnetos that incorporate impulse couplings are used in most engines that require hand cranking, such as those for light planes. Larger engines are usually provided with an auxilliary source of high-tension current for starting. This may be taken from the battery, if one is available, or from a hand-cranked "booster" magneto.

IGNITION WIRING

Ignition Harness. The collection of wires connecting the distributor to the spark plugs is often called the *ignition harness* or *loom.* The high voltages that it handles pose a number of difficult problems not encountered in ordinary wiring circuits. For one thing, special insulation is necessary. Even special insulation may break down when subjected to a high potential and give a "high-tension short" and yet check out satisfactorily if tested with a low-tension test set. It has been found that the most satisfactory type of insulation for ignition wire is obtained by covering the conductor with a layer of natural rubber which has an exceptionally high dielectric strength. Since the natural rubber would deteriorate very rapidly if exposed to oil or fuel vapors or ozone, it is usually covered with a layer of some synthetic such as neoprene. This protective layer is then covered

with an impermeable varnish having a high dielectric strength. The varnish adds to the total value of the insulation and, even more important, prevents vapors or ozone from penetrating through the neoprene to the natural rubber. This is necessary because, although the neoprene is not seriously affected by hydrocarbons or ozone, it is moderately permeable to them, and is not by itself adequate protection for the layer of natural rubber.

The resistance to ozone attack of any synthetic material to be used in the ignition system is very important because the high voltages in the secondary circuit induce a considerable amount of *corona*, a high-tension electrical discharge to the air at very low rates. Such a discharge generates ozone, which, as an active oxidant, attacks natural rubber and many of the synthetics.

Radio Shielding. The use of radio in passenger cars, military vehicles, and aircraft has made it essential to *shield* the ignition system to minimize the high-frequency radiation from that source and hence the amount of interference with radio reception. Several methods have been employed to accomplish this. The conductors for an automobile engine may be placed inside a metal tube that extends from the distributor along the cylinder head past the spark plugs. The individual leads to the various plugs may be carried out through holes in the sides of the tube so that they are unshielded for just the short distance between the tube and the plug. Such an arrangement is simple, inexpensive, and fairly effective in reducing radio interference. Where a much better shielding job is required for aircraft applications, flexible metal hose is employed around each spark-plug lead. Special spark plugs fitted with integral metal tubes around the insulator and special connectors between the plugs and the leads are employed to obtain complete shielding of the high-tension conducting elements. A third method of handling the problem has been introduced recently. This makes use of an outer layer of conducting synthetic rubber on the ignition wire. While the conductivity of the conducting rubber is low, it is sufficient to make the layer effective as a shield.

SPARK PLUGS

Spark-plug Construction. While spark plugs are quite simple in construction, the problems encountered in their development have been many and difficult. These problems may be better understood by referring to Fig. 122, which shows a section through a spark plug for a passenger-car engine. The steel shell has presented few difficulties other than those posed by the standardization of the threads on the lower portion that screws into the cylinder head. Practically all automobile and truck engines take 14-mm spark plugs, while some heavy-duty and medium- or large-sized aircraft engines take 18-mm spark plugs. Some passenger-car engines utilize 10-mm spark plugs because of their lower cost. The 10, 14, and 18 mm refers to the diameter of one of a series of standard metric threads.

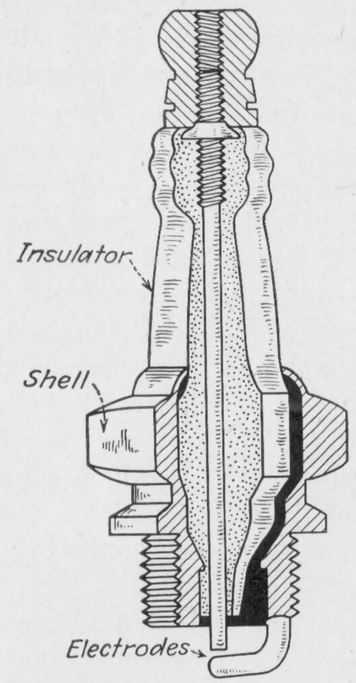


FIG. 122. Section through an automotive spark plug showing its construction.

Insulators. The principal sources of trouble in spark plugs are the electrodes and the insulators. While the problems associated with these elements are interrelated, those of the latter may be conveniently considered first. The insulator of a spark plug is subject to cracking from both high cylinder pressures and mechanical or thermal shock. Insulators may also be seriously affected by moisture or by surface deposits. Trouble was encountered with many ceramic insulators when tetraethyl lead was introduced in fuels because the lead tended to combine with the glaze on the ceramic to form a low-melting-point glass that would drip down on the electrodes and block the spark gap. While many materials including mica have been tried for spark-plug insulators, the most successful have been certain ceramic materials, particularly those made partly or entirely of aluminum oxide.

Electrodes. Theoretically, the optimum shape for the spark-plug electrodes is a sharp point, but a compromise is reached in practice to give an electrode that is not too subject to erosion and corrosion and yet is of as inexpensive a material as possible. Some conception of the severity of the conditions under which the electrodes must operate is given by the fact that electrode temperatures may reach 1200°F. Since this is a dull red heat, corrosion and erosion of the material take place very rapidly. The electrodes are usually made of a high-nickel alloy to withstand these conditions. In a few instances where especially severe conditions have been encountered it has been necessary to use platinum to obtain a satisfactory spark-plug service life.

The corrosion-resistant-alloy grounding electrodes are welded

directly to the steel spark-plug shell, while a short alloy section is welded to a copper center electrode in such a way that the ceramic insulator shields the copper and only the alloy is exposed to the corroding gases in the cylinder. One method often employed to reduce the rate of erosion is to use a single center electrode with two, three, or four grounding electrodes. The spark then tends to jump across the smallest gap and in that way tends to use first one gap and then another, eroding them all uniformly.

Fouling and Preignition. Spark-plug problems would be much simpler if there were not so wide a range of engine requirements. A spark plug whose electrodes run at satisfactory temperatures at moderate speeds and loads may run so cool under idling conditions that portions of the small amounts of oil thrown up past the piston rings by inertia each exhaust-and-intake stroke will deposit on the electrodes and prevent the plug from firing properly. This is called *spark-plug fouling*. On the other hand, the electrodes of the same spark plug may operate at so high a temperature under high engine-output conditions that they will act as a hot bulb and cause preignition. Only by very careful design of the spark plug as a whole is it possible to obtain a plug that operates satisfactorily under all the conditions at which a particular engine may normally be operated. The materials for the insulator and the center electrode as well as the cement sealing the joint between the two must be selected to give just the right combination of heat conductivities and coefficients of thermal expansion so that when the parts are properly proportioned a satisfactory plug can be obtained. Since engines differ in their spark-plug requirements, each manufacturer makes a series of plugs, those spark plugs tending to run hot being referred to as *hot* plugs, while those at the opposite end of the scale are called *cool* plugs. The engine manufacturer may then select a spark-plug type that is best suited to his engine.

The difference between hot and cold spark plugs is most often obtained by varying the length and area of the heat flow path from the tip of the center electrode through the insulator to the shell. The length of the insulator between the tip and the seat in the shell, the amount of gasket surface area, the length and area of the heat flow path through the shell from the insulator to the cylinder head, and other similar factors all have their effects. The appearance of the spark plug after running gives a good indication of how well the plug is matched to the engine. If a white deposit appears on the electrodes, the plug has been running too hot. A sooty deposit indicates

that it has been too cool, while a reddish deposit indicates that it has been running at the proper temperature. The black deposits are carbon, while the lighter colored deposits are lead compounds, the composition and hence the color of which depend on the operating temperature of the plug.

AUTOMATIC SPARK ADVANCE

The effects of spark advance on both knocking combustion and the indicator diagram have been discussed in earlier chapters (pages

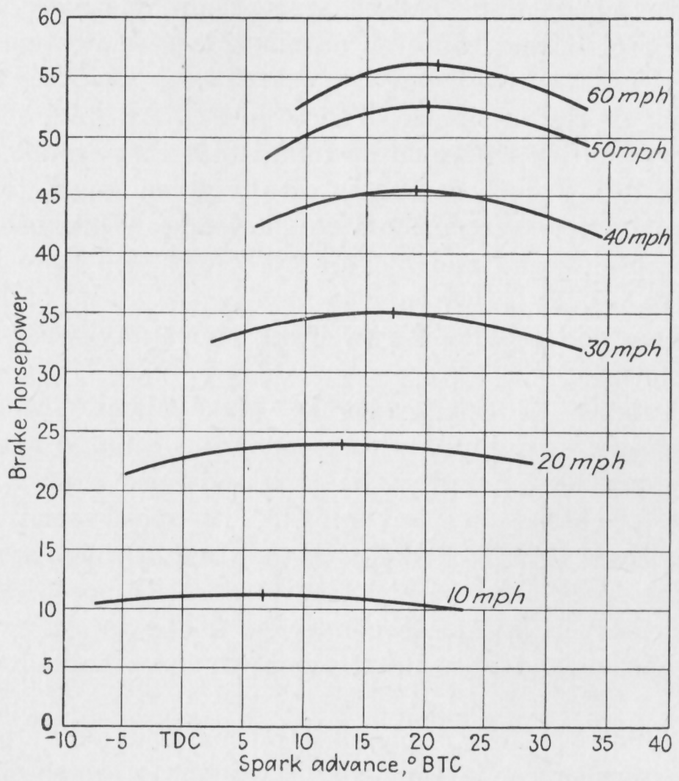


FIG. 123. Effect of spark advance on the full-throttle power output at a series of constant speeds for an automotive engine. (*Hebl and Rendel, Trans. SAE, Vol.* 44, 1939.)

104 and 150). These effects are so marked in terms of motor-vehicle engine performance that it has been found to be very much worth while to provide an automatic spark-advance control for these engines. The following paragraphs indicate the benefits obtainable from such a device together with the characteristics of representative mechanisms.

Effects of RPM and Load on Optimum Advance. The effect of spark advance on engine power output for full-throttle operation at a series of constant equivalent road speeds is shown in Fig. 123.[3] While

the effect of spark advance on output appears greatest at the higher speeds, the *percentage* loss in power with a given change in spark advance from the peak-output condition is practically independent of speed. Since the quantity of fuel and air flowing through the engine is essentially constant for any given curve in Fig. 123, it follows that the spark advance for maximum power is also that for minimum specific fuel consumption.

A curve showing the maximum-power full-load spark advance may be drawn through the peaks of the curves in Fig. 123. It should be noted in this connection that the curves are relatively flat in the peak-power region, and hence an automatic spark advance could be permitted a tolerance of a few degrees without affecting the power appreciably. If curves similar to those in Fig. 123 were obtained for a part-load condition, it would be found that they would be shifted to the right by as much as 10 deg at any given speed, the amount being approximately proportional to the load. While other factors such as spark-plug gap and fuel-air ratio may also have important effects, these usually serve only to modify the shape of the curves indicated here and seldom if ever bring about any reversals in the over-all trends.

Characteristics of Spark-advance Mechanisms. Most motor-vehicle engines are equipped with mechanisms integral with the distributor to provide an automatic spark-advance control that will compensate for changes in speed or load. The speed-sensitive element of these devices is a simple flyball-type unit that, through a cam action, increases the spark advance at a linear rate with speed. The load-sensitive element is a capsule connected to the intake manifold so that an additional advance proportional to the manifold vacuum is provided.

The net effects of these mechanisms on spark advance are shown for a typical passenger car in Fig. 124. Two curves are shown, one for full-throttle and one for road-load conditions. Three other curves are also plotted on the same coordinates to show how closely the actual mechanism comes to meeting ideal requirements. The first of these three curves is that for maximum output at full throttle, while a second is for maximum economy under road-load conditions. Both were replotted from curves such as those in Fig. 123. The third curve shows the maximum spark advance permitted at full throttle by commercial fuels. Such a curve is very well defined for any given fuel, but the differences between different regular gasolines are considerable. Some notion of the magnitude of such differences is given

by Fig. 125, obtained for another representative passenger-car engine. Three types of curve are shown. The first is a single curve for the maximum-power spark advance for full-throttle operation. The second type is a family of curves for trace-knock spark advance (*i.e.*, that for a light constant-intensity knock) for standard reference fuels. Each curve of this family is identified with its equivalent octane number. Three examples of a third type of curve are included in Fig.

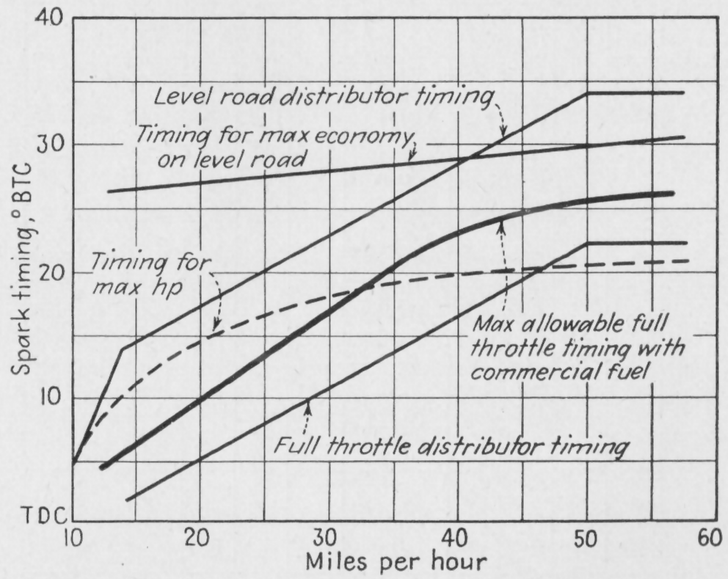


FIG. 124. Curves showing the relation between engine spark-advance requirements under various conditions and the spark advance actually given by the distributor for an automotive engine. (*Hebl and Rendel, Trans. SAE, Vol.* 44, 1939.)

125. These are trace-knock curves representative of commercial gasolines. They differ in shape from the corresponding curves for the reference fuel because they differ in sensitivity to engine conditions, probably because of differences in the types of hydrocarbon structures present in each.

In reviewing Figs. 123 to 125 it is evident that the relatively simple automatic spark-advance devices incorporated in automotive-engine distributors can be adjusted to give a close approximation to the optimum advance over a wide range of operating conditions. Since the curve of power vs. spark advance at a given speed and throttle setting is quite flat, deviations of several degrees from the advance for maximum output and economy give practically negligible losses. The considerable differences between commercial fuels impose both the most important and the most variable limitations on spark timing. In practice, the distributor mechanism is designed to give

approximately the proper shape to the spark-advance curves, and then the distributor may be adjusted on the individual vehicle to give a knock of no more than acceptable intensity under low-speed full-throttle conditions, where knock is most likely to be troublesome.

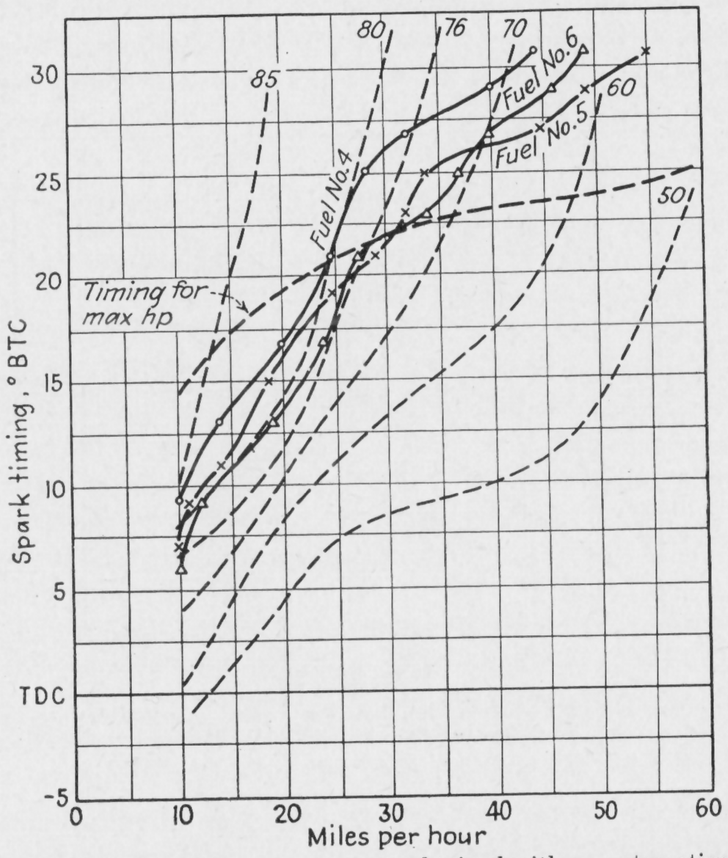


FIG. 125. Trace-knock spark-advance curves obtained with an automotive engine for three commercial gasolines superimposed on a set of trace-knock curves for secondary reference fuels. The octane number of the fuel for each of the latter curves is given at its upper end. (*Hebl and Rendel, Trans. SAE, Vol.* 44, 1939.)

OPERATING PROBLEMS

While experience is the only way in which a person can learn to diagnose and correct ignition trouble on an engine test stand or in a vehicle, a number of common symptoms may be mentioned. The most common causes for misfiring spark plugs are eroded electrodes that give too wide a gap, cracked insulators, or conducting deposits on the insulators. If the gap is too wide, it may be reset by bending the grounding electrode. If the insulator is cracked (the crack may be down inside the shell), the plug must be discarded. Conducting deposits may be in two forms. The first is a deposit of lead compounds and carbon on the portion of the insulator exposed to the combustion chamber. It is most readily removed by sandblasting. The second type of deposit occurs in wet weather on the portion of the insulator outside of the engine. It is particularly likely to occur when the ambient air temperature rises in the early morning hours and a warm rain follows a cold night. Moisture from the saturated air condenses on the cold insulators and, with the accumulation of oil and road dirt already there, provides a conducting deposit. This deposit can be removed simply by wiping the insulator carefully with a dry cloth. Under weather conditions of this sort, moisture may also condense on the high-tension wiring or in the distributor cap and cause one or more high-tension shorts. These may be corrected by putting the car in a warm, dry garage or by putting an electric heater or high-wattage light bulb under the hood in such a way that the resulting heat will dry out the distributor case and the wiring. If this is done, care must be exercised to avoid blistering or burning the insulation on the wiring.

The only other common source of trouble in the ignition system itself is the set of breaker points. It is, of course, always possible that the breaker-point gap may not have been set properly, but this is easily checked. Further, the ignition system is usually not sensitive to small deviations from the prescribed breaker-point gap. However, if any oil or oil vapor gets on these parts, arcing and consequently burning of the point contact surfaces will occur. A short or an open circuit in the condenser will also cause arcing and burning of the points. The two types of trouble can be easily distinguished by the fact that a bad condenser will give a white deposit on the points, whereas oil will leave a black deposit.

Other types of ignition-system trouble are found occasionally. A cracked distributor cap, a short inside the coil, a burned-out coil, a worn breaker-point cam, sticking advance mechanisms, etc., may be encountered but are relatively rare.

Firing Order. In ignition-system maintenance it is often necessary to work with cylinder firing orders. These can be best understood if viewed in a fundamental fashion. The sequence in which cylinders of an engine fire is one of its important design features. It is selected to give as uniform a rate of torque input to the crankshaft as possible and hence is a function of cylinder number and arrangement as well as the arrangement of the crank throws on the crankshaft. This crank-throw arrangement in turn, is selected to give as good an inertia balance of the reciprocating parts as possible. Since

engine dynamics and vibration are beyond the scope of this book, suffice it to say that the firing order is selected to give as smooth-running an engine as possible. While two or more different firing orders may be equally good for a given number and arrangement of cylinders (see Table 12), the firing order of any particular engine cannot be altered without also changing the camshaft. In general, where there is a choice in firing order, the engine designer selects that which will give the least tendency to "wind up" the crankshaft for a portion of each cycle and then allow it to "unwind" during the balance. Take for example the firing orders given in Table 12 for a six-cylinder in-line engine. The 1-2-4-6-5-3 would probably be the worst offender in this respect, whereas the 1-5-3-6-2-4 firing order would be the best because it gives firing impulses that occur first at one end of the crankshaft and then at the other.

TABLE 12. POSSIBLE FOUR-STROKE-CYCLE FIRING ORDERS FOR COMMON ENGINE TYPES

Cylinder arrangement	Number of cylinders	Cylinder firing order
In-line	4	1-2-4-3 or 1-3-4-2
In-line	6	1-3-5-6-4-2, 1-4-2-6-3-5, 1-2-4-6-5-3, or 1-5-3-6-2-4
Horizontally opposed	4	1-3-2-4 1-4-2-3
Radial	7	1-3-5-7-2-4-6
Radial	9	1-3-5-7-9-2-4-6-8

References

1. PETERS, M. F., W. L. SUMMERVILLE, and M. DAVIS: An Investigation of the Effectiveness of Ignition Sparks, *NACA Tech. Rept.* 359, 1930.
2. HARTZELL, H. L.: Effect of Post-war Automotive Practices on Ignition Performance, *Trans. SAE*, Vol. 53, 1945.
3. HEBL, L. E., and T. B. RENDEL: Spark Timing—Its Relation to Road Octane Numbers and Performance, *Trans. SAE*, Vol. 44, 1939.

Problems

1. Sketch a battery-coil ignition system, and label the following: ignition switch, coil, battery, distributor, breaker points, and condenser. Also differentiate between high- and low-tension leads.

2. Determine the gear ratio necessary between the engine and the magneto for (*a*) a two-pole rotating-magnet magneto on a four-cylinder four-cycle engine; (*b*) a four-pole rotating magnet on a nine-cylinder radial engine.

3. Determine the possible firing orders for an eight-cylinder in-line engine in which the various crank throws reach top center in the following order: 1 and 8, 2 and 7, 4 and 5, 3 and 6. Indicate the firing order that would probably give the least difficulty with torsional vibration of the crankshaft.

4. What is a high-tension short?

5. Under what conditions might the following troubles arise?

Oil fouling of the spark plugs
High-tension shorts in the wiring
High-tension shorts across the spark-plug insulation
Burned breaker points (black deposit)
Burned breaker points (white deposit)

CHAPTER VIII

FUELS AND LUBRICANTS

The performance possibilities of an engine are largely determined by the fuel on which it was designed to operate. Similarly, its service life depends to a great extent upon the lubricant used. The sources and the methods of production of fuels and lubricants coupled with their costs, characteristics, effects on engine performance, and the tests required in specifications are all of importance to engineers interested in any type of internal-combustion engine. This chapter is intended to serve as an introduction to these problems.

TYPES OF FUELS AND LUBRICANTS

Sources of Fuels. The reasons for the almost universal use of petroleum as a fuel in the internal-combustion-engine field soon become evident if all potential sources of fuels are examined and the fuel costs and characteristics are compared. Table 13 presents data on representative fuels, including their approximate cost (not including taxes) if purchased in quantities of about 50 gal or the most nearly equivalent standard unit of measure for that fuel. The prices of these fuels, of course, vary considerably from one section of the country to another, as well as with the price structure as a whole. Note the favorable position of gasoline with respect to the other fuels, keeping in mind that gasoline is a highly refined product that can be used to much better advantage in a small, light power plant than a fuel such as coal. Not only does it have a higher heating value and a negligible ash content, but it is a liquid. Liquids are much easier to handle in transferring the fuel from one storage tank to another, and they can be delivered from the tank to the engine more easily and metered far more accurately with relatively inexpensive apparatus than would be possible with a solid fuel.

An examination of Table 13 discloses that all the liquids which appear economically attractive as fuels come from coal, petroleum, or natural gas. Such fuels as alcohol, acetone, or lard oil derived from vegetable or animal matter are characterized by a high cost.

Commercial liquid fuels can be synthesized from coal by several different processes.[1] This was first demonstrated by Germany prior

TABLE 13. HEATING VALUE AND COST OF SOME POTENTIAL FUELS (IN 1947)*

Liquid fuels

Fuel	Density, lb/gal	Lower heating value		Fuel cost, cents/gal	Heat cost, cents/1000 Btu
		Btu/gal	Btu/lb		
Gasoline, 100-octane aviation	6.0	114,000	19,000	24	0.210
Gasoline, premium automobile	6.0	114,000	19,000	13.0	0.114
Gasoline, regular automobile	6.0	114,000	19,000	11.0	0.096
Kerosene	6.59	123,000	18,650	10.3	0.084
No. 1 fuel oil (distillate)	6.91	128,000	18,500	9.3	0.073
No. 2 fuel oil (distillate)	7.07	130,000	18,400	9.3	0.072
No. 3 fuel oil (distillate)	7.30	134,000	18,300	8.8	0.066
No. 4 fuel oil (distillate)	7.48	136,000	18,150	8.5	0.062
No. 5 fuel oil (bunker B)	7.78	140,000	18,000	7.7	0.055
No. 6 fuel oil (bunker C)	8.04	144,000	17,900	7.0	0.048
Arkansaw crude	7.24	132,000	18,250	3.0	0.023
Benzol	7.34	126,000	17,200	21	0.166
Wood alcohol	6.62	56,000	8,400	36	0.642
Grain alcohol	6.58	76,000	11,600	72	0.948
Acetone	6.61	79,400	12,000	55	0.693
Lard oil	7.63	130,000	17,100	110	0.847

Gaseous fuels

Fuel	Specific vol, ft^3/lb at 14.7 psia and 60°F	Lower heating, value, Btu/1,000 ft^3	Fuel cost per 1,000 ft^3	Heat cost, cents/1000 Btu
Hydrogen	189	274,000	$10.00	3.65
Propane	8.6	2,340,000	7.00	0.299
Natural gas	21.4	1,000,000	0.60	1.060
Acetylene	14.57	1,436,000	21.00	1.46

Solid fuels

Fuel	Lower heating value, Btu/lb	Fuel cost per ton	Heat cost, cents/1000 Btu
Bituminous coal	13,000	$13.00	0.050
Anthracite coal	12,500	16.00	0.064
Wood, yellow pine	8,840	12.00	0.068
Charcoal	12,850	55.00	0.214
Corn	8,160	78.00	0.428

* These values are of necessity approximate and vary with the locality and market prices.

to and during the Second World War, when more than half of its gasoline and diesel fuel was derived synthetically from coal. Similar processes are coming into use in the United States to produce fuels essentially similar to petroleum products. While their cost has been higher than that of corresponding petroleum products, our petroleum reserves are dwindling rapidly so that coal must soon become an important if not a major source of liquid fuel. Petroleum derivatives and the similar liquid fuels synthesized from coal have characteristics that are equal to or better than those of any other commercial liquid fuels yet tried in internal-combustion engines.

The differences in heating value of the fuels in Table 13 arise either from differences in their average hydrogen-carbon ratios (hydrogen has roughly three times the heating value of carbon per pound) or from the fact that they are already partly oxidized, *e.g.*, the alcohols or ethers. While the heat of formation of the hydrocarbon itself is also a factor, it is responsible for little variation in fuel heating value except when the compounds in question contain elements other than hydrogen and carbon, *i.e.*, oxygen, nitrogen, etc.

Sources of Lubricants. The situation with regard to potential sources of lubricants is less well defined. While animal and vegetable oils have many desirable characteristics as lubricants, they tend to oxidize and form undesirable deposits to a far greater degree than properly refined petroleum lubricants and hence have disappeared from the combustion-engine field except in greases. This has left petroleum lubricants as the only naturally occurring materials suitable for engine use. Lubricants similar to petroleum derivatives in their composition and characteristics were synthesized from coal on a commercial scale in Germany during the Second World War.[2] In addition, as a result of research carried out prior to and during the Second World War, petroleum-type oils have found serious competitors in the form of certain synthesized compounds, such as the glycols and silicones, that have been found to be exceptionally stable as engine lubricants.[3–5] Since the cost of the lubricant is relatively small as compared with the cost of completely overhauling an engine, the higher cost of some of these synthetic lubricants may be more than offset by lower engine oil consumption, elimination of oil changes, and longer periods between engine overhauls.

STRUCTURE OF HYDROCARBONS

Most of the fuels listed in Table 13 consist of mixtures of hydrocarbon compounds each of which has certain distinct characteristics

of its own. The differences in the ease with which the various types of hydrocarbons ignite and their tendencies to detonate, for example, were pointed out in Chap. IV. Most of the many thousands of hydrocarbons that may be present in petroleum fuels may be classified into one of five groups, viz., normal paraffins, isoparaffins, olefins, naphthenes, and aromatics. Members of each of these groups have related characteristics not only physically and chemically but also in the way they behave in an engine cylinder.[6]

Normal Paraffins. Normal, or *straight-chain*, paraffins, commonly referred to as *n-paraffins*, are a set of chemically stable hydro-

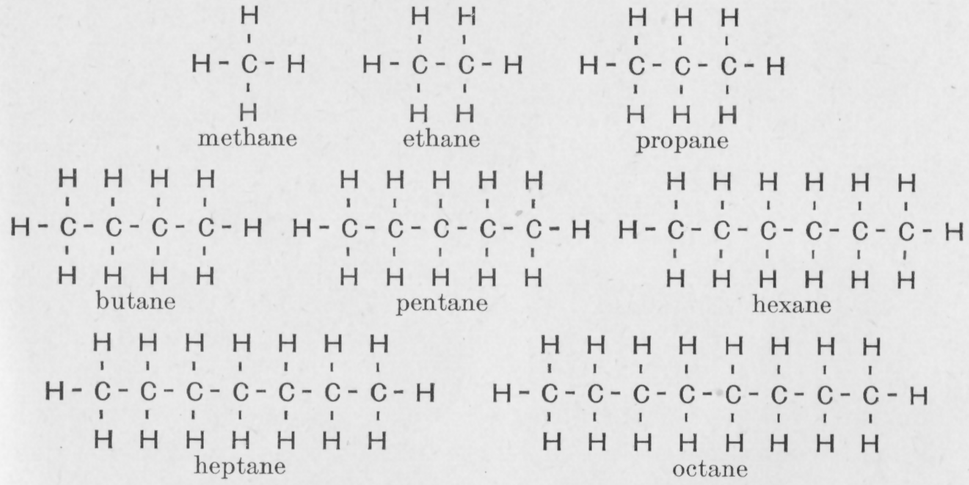


FIG. 126. Structure of the first eight normal paraffins.

carbons having the general formula C_nH_{2n+2}. The structure of the first eight members of this family is indicated in Fig. 126. The actual structure as indicated by X-ray diffraction patterns is really three-dimensional so that the true arrangement, though similar to that of Fig. 126, is in the form shown in Fig. 127. With the exception of the first four members of the family, the name of each compound incorporates the Greek word for the number of carbon atoms in each molecule. This is evident in Fig. 126.

Isoparaffins. An inspection of Figs. 126 and 127 will show that, for paraffins having four or more carbon atoms, other arrangements of the carbon atoms in a molecule are possible without altering the chemical composition. Figure 128 shows the possible structures for paraffins having four and five carbon atoms. Compounds having the same number of carbon and hydrogen atoms but different structures are called *isomers*—hence the term *isoparaffins*. The various isomers may be designated quite specifically by assigning numbers in sequence

to the carbon atoms in the straight chain and referring to the compound by using the name of the straight chain coupled with the numbers of the carbon atoms to which hydrocarbon groups are attached together with the names of those groups. Table 14 shows the various isoheptanes with the names that precisely describe each compound.

The many different isoparaffins are very difficult to separate from both normal paraffins and each other because of the great similarity

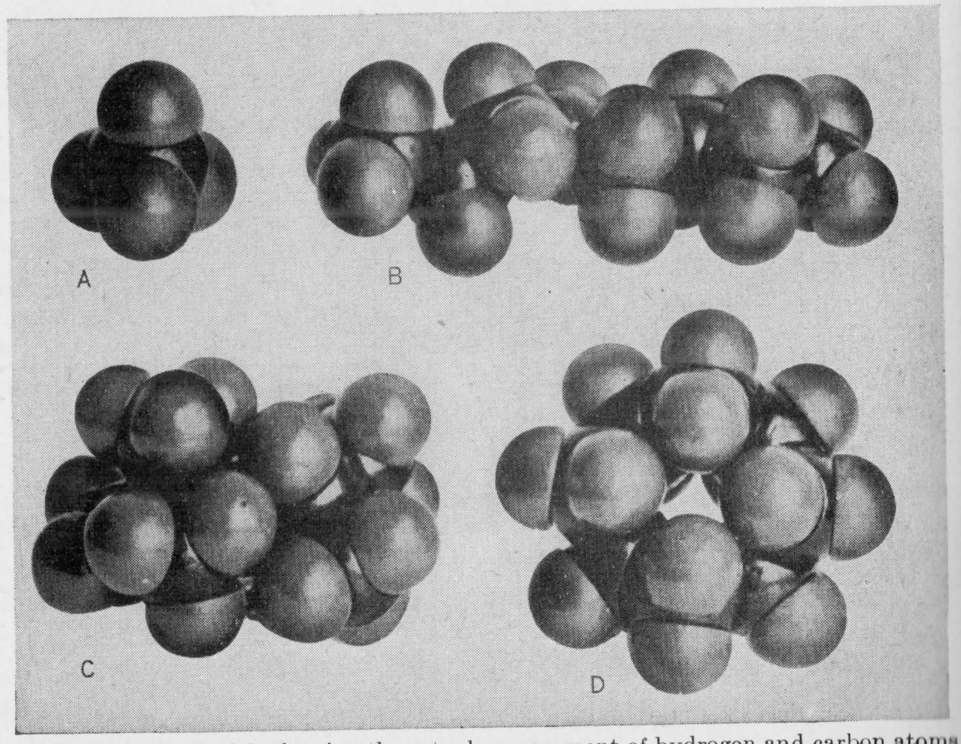


FIG. 127. Scale models showing the actual arrangement of hydrogen and carbon atoms in: (*A*) methane, (*B*) *n*-octane, (*C*) 2, 2, 4-trimethylpentane, (*D*) cyclo-octane.

in their physical and chemical characteristics. Table 14 gives some of the points of both similarity and difference between the isomers of heptane.

Olefins. Olefins are *unsaturated* paraffins; *i.e.*, they are paraffins from which hydrogen atoms have been removed to leave double carbon-to-carbon bonds. This happens in *cracking* operations in which long straight-chain paraffins are broken down to give shorter chain molecules having lower boiling points. Figure 129 shows the structural changes that may accompany such a cracking operation. The names of the individual olefins have been given the suffix *ene* in contrast with the suffix *ane* for the corresponding paraffins. Olefins are

less stable than the paraffins and tend to oxidize to form gums if stored or exposed to light for considerable periods of time.

Naphthenes. Naphthenes are sometimes referred to as *cyclopa-raffins* because they are essentially straight-chain paraffins that have

TABLE 14. STRUCTURE AND PROPERTIES OF ISOMERIC HEPTANES[6]

Name	Structure	Specific gravity 20/4, °C	Boiling point, °C	Melting point, °C	Knock rating*
n-Heptane	C—C—C—C—C—C—C	0.68378	98.42	− 90.6	0
2-Methylhexane	C—C—C—C—C—C C	0.67869	90.1	−118.2	45 m
3-Methylhexane	C—C—C—C—C—C C	0.6870	91.96	−119.4	65 r
3-Ethylpentane	C—C—C—C—C C C	0.69818	93.47	−118.6	62 r
2,2-Dimethylpentane	C C—C—C—C—C C				
3,3-Dimethylpentane	C C—C—C—C—C C	0.67388	79.3	−124.0	93 m
2,3-Dimethylpentane	C—C—C—C—C C C	0.69330	86.1	−135.0	84 m
2,4-Dimethylpentane	C—C—C—C—C C C	0.69514	89.8	.....	89 m
2,2,3-Trimethylbutane	C C—C—C—C C C	0.67275	80.6	−119.1	82 m
		0.69000	80.87	−25.06	100+

* The letter "r" or "m" following the numerical octane rating denotes the method of determination, *i.e.*, the CFR *motor* or *research* methods.

had the ends of each chain joined together to form rings such as those in Fig. 130. These compounds have the same characteristic formula as olefins, viz., C_nH_{2n}, but are more like paraffins in their chemical stability.

Aromatics. The benzene ring is the basic structural element in a large group of hydrocarbons called *aromatics*. The term was first

applied because many of these compounds have very penetrating and distinctive odors. Figure 131 gives diagrams for the structures of a number of aromatic molecules. Unlike the hydrocarbons previously discussed, the carbon atoms in the actual benzene ring in the aromatics lie in one plane. The double carbon-to-carbon bond shown in the representation of Fig. 131 does not indicate unsaturation, for the benzene ring is very stable and is not inclined to take on oxygen atoms to take the place of the double bonds between carbon atoms. The precise nature of the benzene-ring double bonds is too complex to be presented here.

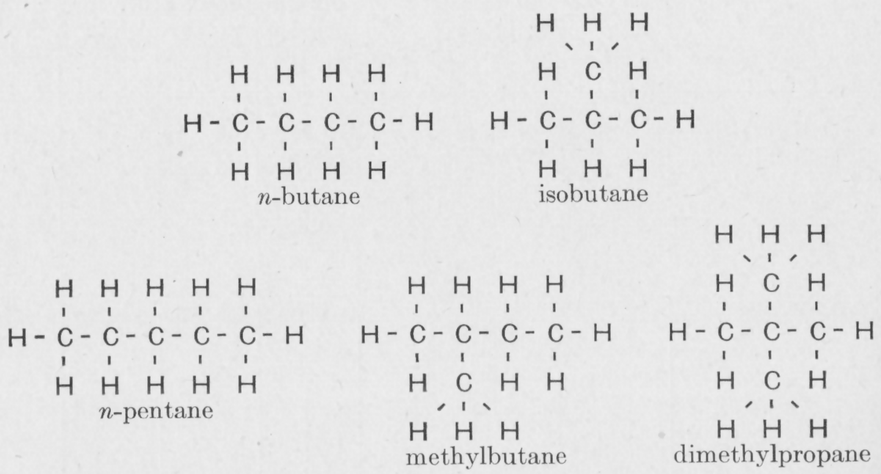


FIG. 128. Structure of the possible isomeric forms of butane and pentane.

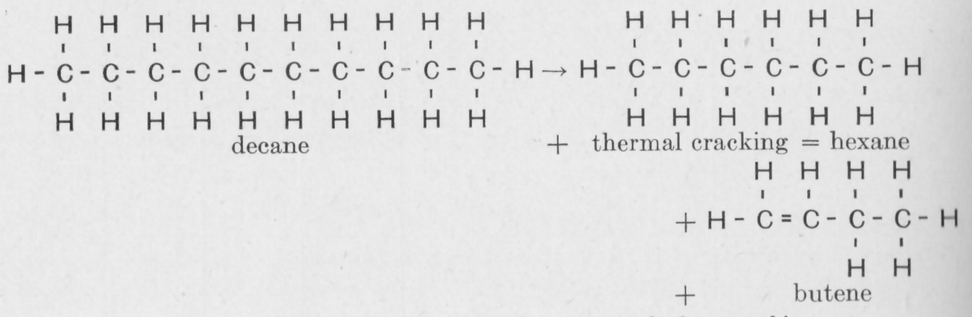


FIG. 129. Hydrocarbon structures before and after cracking.

Properties of Representative Hydrocarbons. The data in Table 15 give the more important properties of typical compounds in petroleum fuels. These data are more representative than might at first appear, for the boiling range of commercial gasoline and high-grade diesel fuels limits the number of hydrocarbons likely to be present in appreciable quantities to only a few thousand. Note that data are also given for compounds such as methanol, which, though not ordinarily used in gasoline, is of interest as a fuel.

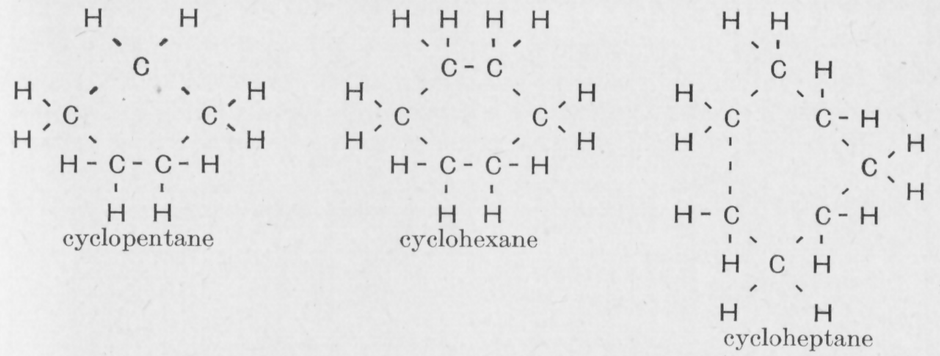


FIG. 130. Structure of typical naphthenes.

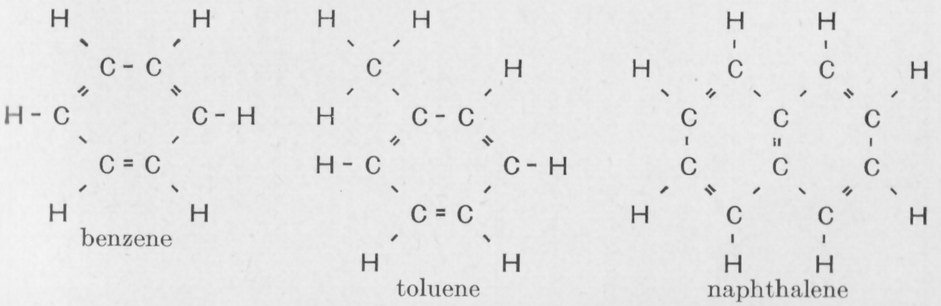


FIG. 131. Structure of typical aromatic hydrocarbons.

FUEL-REFINING PROCESSES

The modern petroleum refinery is a complex chemical-processing plant employing many types of process and equipment. Figure 132 gives a simplified picture of the flow of crude oil and intermediate and finished products through a refinery, together with the types of processing employed.[8] Crude oil entering the refinery is first fed to a still in which the more volatile components are distilled off and separated according to their boiling ranges in a fractionating tower. Gases such as propane and butane not liqueified in the fractionating tower may be compressed and liquefied for temporary storage, or they may be used directly as basic materials for making high-octane gasoline or synthetic rubber. They are valuable for use as "bottled gas" and for increasing the volatility of winter-grade gasoline. The lighter distillates having a boiling-point range from 100 to 430°F form the base stock for motor gasoline, a part of the next heavier distillates ranging from 380 to 540°F are further refined and sold as kerosene and the next heavier as the lighter fuel oils, while the heaviest distillates form the base stock for the light and medium lubricants. No attempt is

TABLE 15. PROPERTIES OF REPRESENTATIVE COMPOUNDS IN HYDROCARBON FUELS[6,7]*

	Mol formula	Mol wt	Melting point at 14.7 psia, °F	Boiling point at 14.7 psia, °F	Density of liquid: Specific gravity at 60/60, °F	Density of liquid: °API	Lb/gal at 60°F	Specific heat 14.7 psia: C_p vapor, Btu/(lb)(°F)	Specific heat 14.7 psia: C_v vapor, Btu/(lb)(°F)	Specific heat 14.7 psia: C_p liquid, Btu/(lb)(°F)	Heat of vaporization 14.7 psia, Btu/lb at boiling point	Ignition temperature, °F	Ft³ air to burn 1 ft³ gas	Combustibility limits: Upper F/A	Combustibility limits: Lower F/A	Lower heating value: Btu/gal liquid at 60°F	Lower heating value: Btu/lb	Ratio gas vol per liquid vol	Specific gravity (air = 1.00)	Lb/gal at 60°F	Density of liquid: °API	Density of liquid: Specific gravity at 60/60, °F	Heat of vaporization 14.7 psia, Btu/lb at boiling point	Specific heat 14.7 psia: C_p vapor, Btu/(lb)(°F)	Specific heat 14.7 psia: C_v vapor, Btu/(lb)(°F)	Specific heat 14.7 psia: C_p liquid, Btu/(lb)(°F)	Octane rating	Octane rating	Cetane rating
Hydrogen	H_2	2.016		−423							192	1076	2.38	0.172	0.029		43,000												
Carbon	C	12.010															14,600												
Methane	CH_4	16.042	−296.5	−258.5	0.3	340	2.5	0.554	0.402		245	1346	9.55	0.088	0.026		21,597		443										
Ethane	C_2H_6	30.068	−297.8	−128.2	0.374	247	3.11	1.038	0.415	0.347	211	1050	16.71	0.131	0.031		20,597		294.5										
Propane	C_3H_8	44.094	−305.9	− 43.8	0.508	147.0	4.23	1.522	0.390	0.346	183	995	23.87	0.131	0.031		20,015		272.7										
n-Butane	C_4H_{10}	58.120	−216.9	+ 31.1	0.584	110.8	4.86	2.006	0.396	0.363	166	961	31.03	0.155	0.030		19,795		237.8										
n-Pentane	C_5H_{12}	72.146	−201.5	97.0	0.631	92.7	5.25	2.491	0.402	0.376	153	933	38.19	0.172	0.008	101,000	19,745		207.0										
n-Hexane	C_6H_{14}	86.172	−139.7	155.7	0.664	81.6	5.53	2.975	0.406	0.384	146	909	45.35	0.035			19,432		182.4										
n-Heptane	C_7H_{16}	100.198	−131.1	209.1	0.688	74.2	5.73	3.459	0.415	0.397	138	893	52.06	0.165	0.033		19,408		162.5										
n-Octane	C_8H_{18}	114.224	− 70.0	258.1	0.706	68.5	5.89	3.943	0.420	0.404	131	880	59.67		0.038		19,329		146.5										
n-Nonane	C_9H_{20}	128.250	− 64.7	303.3	0.722	64.5	6.01	4.428	0.429	0.415	125	871	66.85		0.037		19,292		133.2										
n-Decane	$C_{10}H_{22}$	142.276	− 21.5	345.2	0.734	61.5	6.11	4.912	0.436	0.424	120	866	73.99		0.033		19,205		122.1										
n-Cetane	$C_{16}H_{34}$	226.4	65	536	0.774	51.5	6.43					390					19,091												
Pentatriacontane	$C_{35}H_{72}$	492.3	176	628	0.781	49.5	6.51										19,032												
Isopentane, or 2-methylbutane	C_5H_{12}	72.1	−256	82	0.613	99.0	5.17				146				0.034	100,000	19,597										90		
Triptane, or 2,2,3-trimethylbutane	C_7H_{16}	100.2	− 13	178	0.690	72.9	5.76				125					110,000	19,100										140		
Isooctane, or 2,2,4-trimethylpentane	C_8H_{18}	114.2	−162	211	0.692	73.5	5.77				117	1350				110,000	19,159										100		
n-Pentene	C_5H_{10}	70.1	−230	86	0.642	87.0	5.39										18,034										80		
n-Octene	C_8H_{16}	112.2	−152	251	0.751						132						19,190										34		
Cyclopentane	C_5H_{10}	70.1	−138	121	0.746	56.7	6.22										17,494										82		
Cyclohexane	C_6H_{12}	84.1	44	177	0.778	51.6	6.50				156						17,494										77		
Benzene	C_6H_6	78.1	42	176	0.88	29.0	7.34				169	1364	35.7	0.234	0.038	126,000	18,800										97		−10
Toluene	C_7H_8	92.1	−139	231	0.87	31.0	7.23				156	1490	42.9	0.230	0.054	126,000	17,190										104		−21
Naphthalene	$C_{10}H_8$	128.2	177	424	0.975								57.1			126,000	17,460												
Cumene, or isopropyl benzene	C_9H_{12}	120.2	−141	306	0.862	32.6	7.19				134					128,000	16,730										78		
Aniline	$C_6H_5NH_2$	93.1	+ 21	364	1.022		8.53				187					128,000	17,700												
2,4-Xylidine	$C_6H_3NH_2(CH_3)_2$	121.2		420	0.974	13.8	8.13				150					128,000	15,000												
α-methyl naphthalene	$C_{11}H_{10}$	142.2	− 7.6	470	1.025											128,000	15,700										0		
Tetraethyl lead	$PB(C_2H_5)_4$	323.4	−213	360	1.653		13.8				73																		
Ethylene dibromide	$C_2H_4Br_2$	187.8	+ 50	269	2.181	46.4	18.2				82																		
Methanol	CH_3OH	32.0	−144	149	0.792		6.62				502						8,644										98		
Ethanol	C_2H_5OH	46.0	−179	172	0.785	47.1	6.58				396					56,000	11,604										99		185
Ether	$(C_2H_5)_2O$	74.08	−177	94	0.714											76,000													
Diethyl-dioxyl ether				340	0.91																								
Acetaldehyde	CH_3CHO	44.0	−190	68	0.781																								

* Much of the data in this table came in the form of isolated fragments from dozens of widely scattered papers, most of them in the *SAE Transactions* or the *Petroleum Refiner*.

made to distill over all the crude, for excessive carbon formation would occur in the still. The liquid remaining after the initial distillation is known as the *residuum*. It may be sold as heavy fuel oil but is more often subjected to further processing, some of it going to make the heavier lubricants. Since there is a much larger and more profitable market for gasoline than there is for fuel oil, a part of both

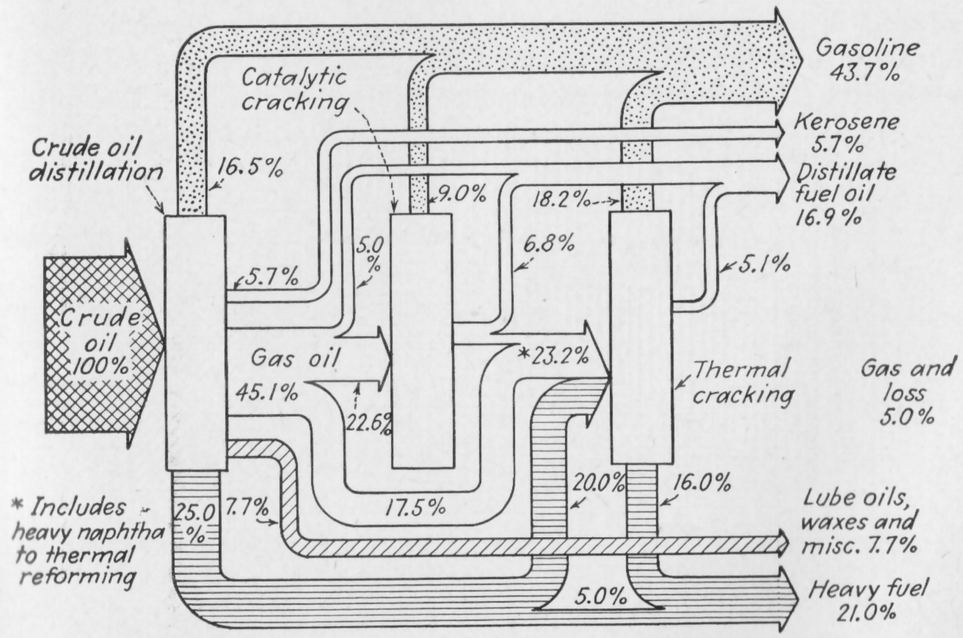


FIG. 132. Flow sheet showing the more important refining processes and the estimated relative amounts of petroleum flowing to each during the course of refining. Values given are for the United States as a whole. (*Holaday and Happel, Trans. SAE, Vol.* 51, 1943.[8])

the heavier distillate and the residuum is fed to cracking units, where the heavy molecules are broken down into the more marketable lighter hydrocarbons. The nature of these and other important processes employed in carrying out refinery operations is indicated in the following sections.

Distillation and Fractionation. Distillation followed by liquefaction in a fractionating tower provides a relatively simple and effective method of separating hydrocarbons having appreciably different boiling points. Motor fuel obtained directly from the crude by distillation is called *straight-run* gasoline. Distillation is also used for separation purposes after other refining processes. If the process is employed to separate the distillate into very narrow boiling-point ranges, it is called *superfractionation*.

Absorption. Natural gas from many wells contains appreciable percentages of light hydrocarbons suitable for gasoline. These components can be separated from the still lighter hydrocarbons that make up the bulk of the gas by compressing the mixture, cooling it, and then bubbling it through light oils into which the heavier gas components readily dissolve. This *absorption* procedure to recover these gases is necessary because the gasoline hydrocarbon vapors are present in the gas in such small percentages that their partial pressures are quite low relative to those of the other components. Refrigeration is often employed to increase the yield from the absorption

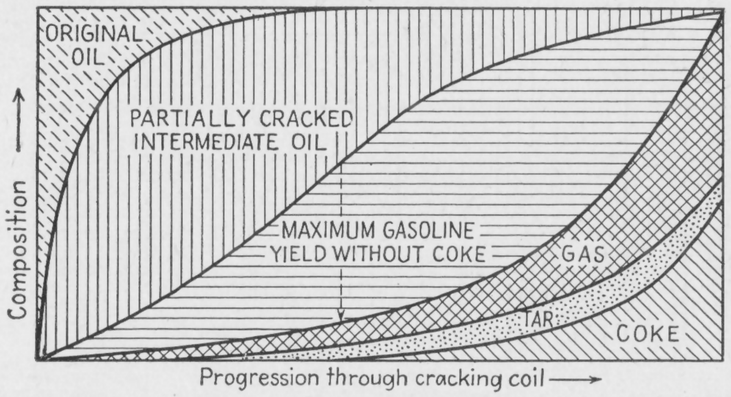


FIG. 133. Effect of temperature on the composition of the products of a thermal cracking unit.

portion of the process. The gasoline absorbed in the light oil can be separated by distillation and the oil recycled through the process. The fuel obtained by liquefaction from natural gas is called *natural* or *casing-head* gasoline.

The absorption process is also employed to recover vapors suitable for gasoline from gases produced in the refinery. Very often the process is carried out with a mixed feed of natural and refinery gases.

Thermal Cracking. As pointed out in the discussion of olefins on page 218, the heavier hydrocarbon molecules can be broken up into two or more fragments to yield a volatile product suitable for motor fuel. One method of accomplishing this is to subject the low-volatile distillates (gas oil) and/or the residuum from the initial distillation to high temperatures and pressures. If the cracking is carried out with the charge in the liquid state, it is called *liquid-phase* cracking, the temperatures required running from 850 to 1000°F and the pressures from 1,000 to 1,500 psi. If carried out in the vapor state, the process is called *vapor-phase* cracking, the temperatures running from 1000 to 1500°F and the pressures from 50 to 150 psi. As shown in

Fig. 133, the yield from a thermal cracking unit depends upon the temperature and pressure conditions; *i.e.* the higher the temperature and pressure, the greater the percentage of gasoline obtained per unit of charging stock. However, since the amount of coke deposited also increases, it imposes an economic limit on the cracking conditions; for not only does the oil going into coke represent a loss, but the coke itself builds up in the form of hard deposits that clog the cracking unit. Removal of such coke deposits represents an added expense.

Catalytic Cracking. Higher gasoline yields with lower losses can be obtained under much milder cracking conditions through the use of any of several catalysts including clay, aluminum hydrosilicate, alumina silica, and nickel oxide. This type of cracking process has come into widespread use in many different forms since the first large-scale units began operation in 1936. By properly adjusting the temperatures, pressures, etc., catalytic cracking processes can be made to yield relatively high percentages of certain hydrocarbon types such as isoparaffins. Because of this, catalytic cracking processes have been the best source of high-octane gasoline.

The Houdry process, the first of the catalytic-cracking processes, is fairly representative. Petroleum gases of higher molecular weight than gasoline are passed through a retort containing a clay catalyst which promotes cracking reactions that produce types of hydrocarbon desirable for gasoline. Some carbon is released by the cracking reactions and tends to clog the bed of catalyst. It must be burned out at intervals with air supplied under a pressure of about 60 psia. To permit continuous operation the retorts are operated in groups of three, two being used for cracking while the third is being decoked. The very large amounts of air and hence power required to burn out the coke and regenerate the catalyst made this a natural application for the gas turbine. Much of the power put into the air in the compressor could be recovered in the turbine along with a part of the heat released by the combustion of the carbon in the catalyst retort. While as a power plant this arrangement had too low a thermal efficiency to make it an attractive source of power, it did afford an economical way of carrying out a chemical process and gave some electric power as a by-product. In the first unit the by-product electric power amounted to about 20 per cent of the power developed in the turbine, making it the first commercial gas-turbine power plant.

Polymerization. The light unsaturated hydrocarbons such as ethylene and butylene derived from cracking processes can be made

to combine with each other to yield olefins of higher molecular weight. These have proved to be excellent blending agents for motor fuels. Polymerization makes it possible to convert gases from refinery processes such as cracking to a high-quality gasoline, a very worth-while operation though at one time these gases were burned in refinery furnaces or just wasted in flares.

Re-forming. The heavier straight-run hydrocarbons in the gasoline and kerosene ranges are low in antiknock quality. They may be subjected to processes similar to those of thermal or catalytic cracking to bring about both some breakdown and an internal rearrangement of the molecules. The product of such a re-forming operation has a lower boiling point than the charging stock and a much improved antiknock quality as a result of the formation of isomeric and cyclic hydrocarbons.

Isomerization, Alkylation, Cyclization, and Aromatization. By proper selection of catalyst and operating conditions, certain specialized types of reaction may be obtained with particular charging stocks. *Isomerization* is the term applied to a process that changes the space structure of a hydrocarbon without changing the number of carbon and hydrogen atoms in the molecule. It has been widely applied to form isopentane from a charging stock consisting primarily of normal pentane. *Alkylation* is the term applied to a process that brings about the combination of an olefin with a paraffin or other saturated hydrocarbon; *e.g.*, butylene plus isobutane can be made to give isooctane. *Cyclization* is the term applied to a process in which the ends of a straight-chain molecule are joined together to form a naphthene, while *aromatization* is similar except that an aromatic hydrocarbon is formed. In both cyclization and aromatization, dehydrogenation takes place, for the hydrogen atoms at the ends of the straight chain must be removed. The hydrogen obtained as a by-product is particularly valuable since it may be used in the same refinery for other processes involving hydrogenation.

Blending. The products of the various refinery operations designed to yield gasoline must be blended together if commercial fuels of the desired characteristics are to be obtained. Some components are either too volatile or not volatile enough, too low in antiknock quality, too high in olefin content (and hence inclined to form gums), etc. The components especially high in antiknock quality may be used in aviation gasolines, while the balance may go into motor fuel. The volatility of each gasoline must be carefully controlled with regard both to the types of application and to the season

of the year. This requires that relatively large percentages of butane, for example, be separated and stored in summer so that it may be added to the gasoline in winter to produce a fuel having good cold-starting characteristics. Tetraethyl lead, by far the most effective of a number of compounds whose presence in gasoline acts to suppress detonation, is usually added as an inexpensive means of greatly improving the antiknock quality of the fuel. Since the antiknock quality of the refinery output tends to vary with the type of crude as well as the relative demand for the various products, the use of varying amounts of tetraethyl lead provides an easy means of maintaining a uniform antiknock quality of finished product.

Oxidation inhibitors may also be added to gasoline during the blending process to prevent slow oxidation of certain types of hydrocarbon, principally the olefins formed in the cracking operations. The olefins have good antiknock qualities but were once regarded unfavorably for use in gasolines because of their tendency to oxidize and form gums. If gums once form, they tend to clog fuel strainers, fuel pumps, and carburetors as well as to form deposits in the intake manifold, on intake valve stems, and sometimes on the pistons and rings. Such deposits are likely to cause intake-valve sticking, ring sticking, or related troubles. However, certain complex compounds have been found that have proved to be very effective in preventing this oxidation even when only very small percentages are added to the gasoline. Since the early 1930's these inhibitors have been widely and successfully used in the products of cracking stills to provide desirable fuels.

FUEL PERFORMANCE RATING

Fuel-rating Scales. The resistance of a fuel to knock is at once one of its most important characteristics for engine use and one of its most difficult properties to determine satisfactorily. Many different reference scales have been proposed, perhaps half a dozen have been fairly widely used, but none has proved entirely satisfactory.[9,10] It has been found necessary to employ reference fuels in all knock-rating procedures because engines differ so widely in their characteristics. The interrelationships of imep, spark advance, compression ratio, combustion-chamber shape, intake-manifold pressure and temperature, fuel-air ratio, cylinder-head-metal temperature, etc., are much too complex to permit the use at any one or any single group of these variables as a reference scale applicable to all types of engine.[10–12] Even the use of a reference fuel to which other fuels may be compared has its disadvantages. It is unfortunate but true that any given fuel

often behaves differently in different types of engine so that it may be better than a reference fuel in one engine and worse in another. Disadvantageous and contradictory as this may appear, this fact in and of itself precludes any more simple knock-rating procedure than one making use of reference fuels. No single reference scale such as one involving compression ratio could possibly admit of or allow for such erratic behavior.

The octane scale was evolved in the middle and latter 1920's and was adopted by the CFR (Cooperating Fuel Research Committee of the SAE and the API) in essentially its present form in 1930. At the time when it was devised, one of the isomers of octane—2,2,4-trimethyl pentane—had just been isolated and had been found to be more resistant to knock than any readily obtainable pure hydrocarbon then known. By mixing isooctane with *n*-heptane (one of the hydrocarbons most prone to knock) it was found that a mixture could be obtained which would match any gasoline then in use. The rating of the fuel was expressed in terms of the percentage of isooctane in the octane-heptane mixture. Thus, the more resistant a fuel to knock, the higher its octane number.

It was mentioned above that, of two fuels having roughly the same resistance to detonation, one might prove superior in one engine, while the other would be the better in another. To avoid irregularities and inconsistencies, a single standard engine model and test procedure were adopted for fuel-test work. The engine was made to be as nearly representative of all types of automotive engine as possible, while the test conditions were made typical of severe service conditions. Several different procedures employing this engine were devised and have remained in use. The most widely used is the CFR motor method, which was originally devised for determining the knock characteristics of motor fuels. It has been adopted by the ASTM and is the method employed for the commercial rating of automotive gasolines. The CFR research method was developed a little earlier than the motor method and is still used for some types of experimental and development work. The CFR aviation method is an outgrowth of a method developed by the United States Army at Wright Field for rating the knock characteristics of aviation fuels. It has been adopted by the ASTM and is the standard method for rating the lean-mixture performance of commercial aviation fuels. The test conditions for each of these three methods are given in Table 16.[13]

Knock intensity is indicated by a bouncing pin in the motor and research methods. The abrupt change in cylinder pressure under knocking conditions acts on a diaphragm mounted directly over the portion of the combustion chamber farthest from the spark plug, *i.e.*, the point where knock will occur. A shock load on the diaphragm causes it to deflect vertically and set a pin bouncing. Carefully adjusted breaker points mounted over the upper end of the pin are set vibrating by the bouncing pin. The intermittent contact of these breaker points completes a d-c electric circuit in which is connected a highly damped milliammeter called a *knockmeter*. The more violent the knock, the greater the current flow and the higher the knockmeter reading.[14]

A thermal plug is used to indicate knock in the aviation-knock-test method. The procedure followed is quite complex as the test is designed to serve as a measure of both the knock and the preignition characteristics of the fuel. In brief, knock or preignition is indicated if the thermal-plug temperature exceeds a certain value specified as normal for the particular conditions of the test.

TABLE 16. OCTANE-RATING METHODS

Item	CFR motor method	CFR research method	CFR aviation method (lean mixture)
Engine speed	900 rpm	600 rpm	1,200 rpm
Coolant jacket, °F	212	212	374
Spark (BTDC)	Varied	13 deg	35 deg
Mixture temperature, °F	300	.....	220
Intake-air temperature, °F	.....	125	125
Oil temperature, °F	.....	.....	150
Knock indication	Knockmeter	Knockmeter	Thermal plug
Reference knockmeter reading	55 ± 3	55 ± 3	

Reference Fuels. The *primary reference fuels* for the octane scale are *n*-heptane and 2,2,4-trimethylpentane. The technical grades of these compounds having a purity of 96 per cent or more are used for this purpose. Since even the technical grades of these compounds are expensive, *secondary reference fuels* are often employed. These fuels are mixtures of hydrocarbons whose knock characteristics have been carefully evaluated in terms of the primary reference fuels.

Nonlinearity of the Octane Scale. A disadvantage of the octane scale, particularly with respect to possible extrapolation for fuels having a greater resistance to knock than isooctane, is its nonlinearity in

terms of any convenient engine operating parameter such as imep, manifold pressure, or compression ratio. The knock-limited performance of a typical engine is shown as a function of octane number in Fig. 134. Two curves are given—one obtained with a constant compression ratio while the manifold pressure was varied, and the other obtained at a constant manifold pressure while the compression ratio was varied. Both show that the addition of a small amount of *n*-heptane to isooctane has a much greater relative effect than the

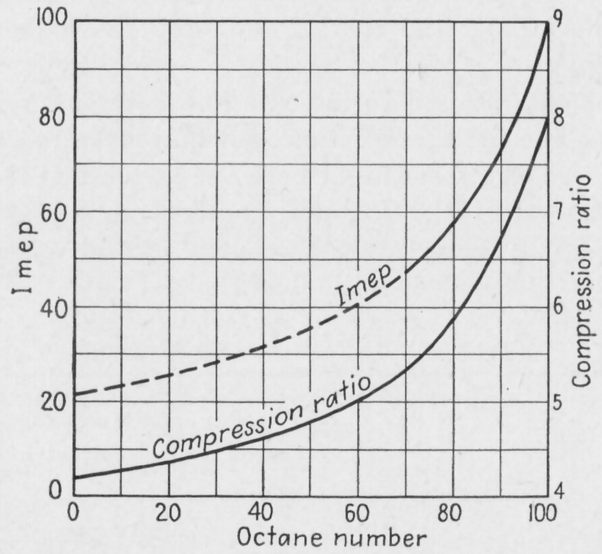


FIG. 134. Effect of octane number on the knock-limited compression ratio for a constant charge inlet pressure, and on the knock-limited imep for a constant compression ratio and a variable charge inlet pressure. (*Brooks, Trans. SAE, Vol.* 54, 1946.[9])

addition of the same amount of isooctane to *n*-heptane. This is in line with the notions advanced in Chap. IV on preflame reactions and the formation of compounds that may serve as chain carriers, for the presence of relatively small amounts of such compounds may serve to start the reaction at an appreciable rate. While Fig. 134 is for mixtures of *n*-heptane and isooctane, the same general relations characterize blends of almost any two fuels having substantially different knock limits.

Fuel Sensitivity. The characteristics of the premium fuels developed for aircraft can be described properly only in a complex fashion. While these characteristics are primarily concerned with aircraft applications, their implications are much broader, especially in the motor-vehicle field, and hence are discussed briefly in this and the following sections.

As shown in Chap. IV, the knock-limited imep of any given engine operated under a generally fixed set of conditions varies with the inlet temperature of the mixture, the fuel-air ratio, the rpm, and the combustion-chamber-wall temperature. The degree of this variation depends upon the particular fuel, some fuels being very sensitive to changes in these quantities, while others are relatively insensitive. In general, the degree of sensitivity of all types of fuel seems to increase quite rapidly with an increase in octane number. In all cases the aromatics tend to be the most sensitive. Fuel sensitivity is important for several reasons. In the first place, a sensitive fuel varies widely in its suitability for different types of engine and thus requires a relatively complex representation of its performance characteristics. In the second place, and this is fundamentally the more important consideration, ultrahigh engine performance appears to be permitted only by rather highly sensitive fuels and then only in engines designed to give very mild temperature conditions, *i.e.*, relatively low intake manifold and combustion-chamber-wall temperatures.[10] Under such conditions, engines have been run without knock at imep's close to 1,000 psi—nearly ten times the highest value ordinarily obtained in automotive engines with regular grade fuels.

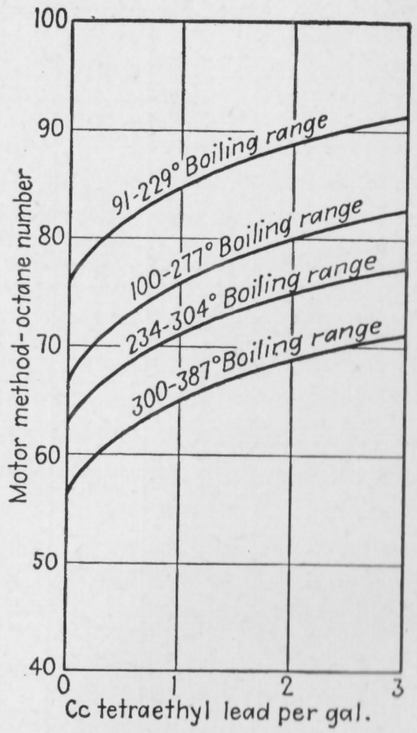


FIG. 135. Tetraethyl lead susceptibilities of narrow boiling-point fractions of a thermally cracked gasoline. (*Holaday and Happel, Trans. SAE, Vol.* 51, 1943.[8])

Rating Fuels of Better than 100 Octane. Since the adoption of isooctane (2,2,4-trimethylpentane) as a basis for fuel rating, fuels have become available that have such superior characteristics that they cannot be rated by the octane scale. Several methods for extrapolating the octane scale have been employed.[9] One method has expressed fuel performance in terms of the equivalent mixture of isooctane and tetraethyl lead, the amount of the latter being given in cubic centimeters per gallon of isooctane. Thus on this basis an aviation gasoline might be rated as equivalent to isooctane plus 3 cc TEL

(tetraethyl lead). This method of extension of the octane scale is adequate only for the region immediately above 100 octane and gives a highly nonlinear extension, as may be deduced from a lead-susceptibility curve such as that in Fig. 135. Other scales have been proposed, the most promising of which at the time of writing appears to be one

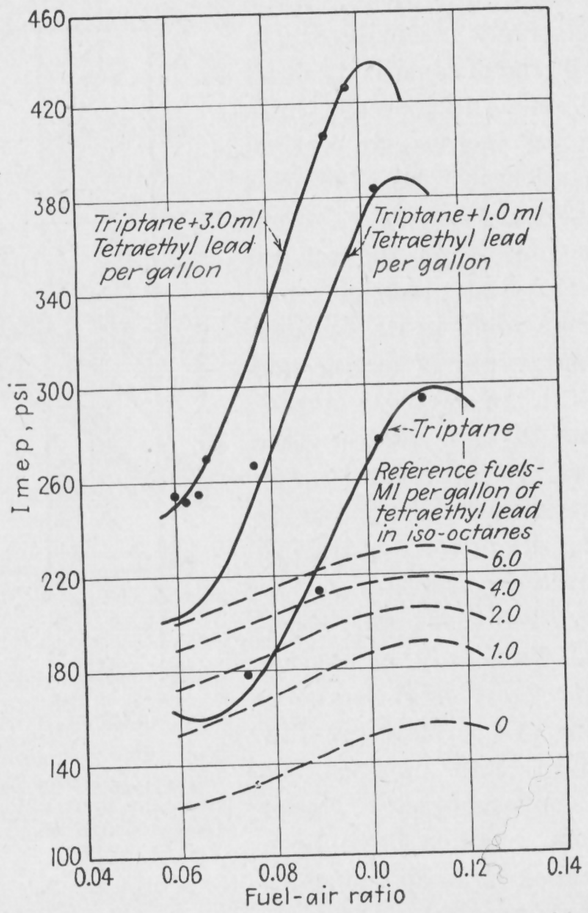


FIG. 136. Mixture susceptibility curves showing the knock-limited imep obtained with isooctane and with triptane. The effects of the addition of tetraethyl lead are also shown. (*Kettering, Trans, SAE, Vol.* 53, 1945.)

employing leaded triptane and heptane as primary reference fuels.[9] Figure 136 shows two sets of *mixture-susceptibility curves*, one run with the octane-scale reference fuels and the other with the leaded triptane-heptane reference fuels. Note the much greater range provided by the latter, as well as the greater sensitivity to mixture strength. As was pointed out in the previous section, the greater sensitivity is characteristic of the so-called "superfuels." It might be mentioned that, in some engines under extremely severe temperature conditions, triptane will actually give poorer performance than isooctane. It is something of a coincidence that isooctane is so insensitive to temperature or fuel-air ratio, for it is much less sensitive than almost any other high-antiknock fuel likely to be used even for test purposes.

Cetane Rating of Diesel Fuels. Performance rating of diesel fuels has received only a small fraction of the attention that has been devoted to the corresponding properties of gasolines. This has been due partly to the smaller demand for diesel fuel, partly to the much smaller potential gain in engine performance from fuel improvements, and partly to the very great effort made by the diesel manufacturers to build engines that would run satisfactorily on the cheapest possible fuels, *i.e.*, by-product materials from the manufacture of gasoline. They have done this because low operating costs have been the biggest single sales feature for their engines, and any attempt to obtain special fuel properties would inevitably mean higher fuel costs. In the automotive diesel field the higher engine performance possible from improved fuels has more than paid for the extra cost, but this has not been true to the same extent for heavy stationary and marine diesels. The more important the factor of engine weight as compared with fuel cost, the more important the fuel characteristics. Because of the importance of this factor of engine weight, the highest levels of fuel development have been reached in aviation gasolines, the next highest in motor-vehicle gasolines, the next in automotive diesel fuels, and the lowest in heavy diesel fuels. In each case the economics of service operation have determined the extent of the development work.

The close correlation between knock and ignition lag in the diesel have led to the use of the latter quantity as a measure of fuel performance. As with gasolines, the blend of two reference fuels that matches the performance (ignition lag) of the test fuel provides the basis for fuel-rating work. The primary reference fuels are cetane, $C_{16}H_{34}$, a normal paraffin having the unusually high ignitability of the higher straight-chain paraffins, and α-methyl naphthalene, an aromatic hydrocarbon of very low ignitability. (The more important physical properties of these two compounds are included in Table 15.) The percentage of cetane in a mixture of the two is taken as its *cetane number*. As in octane-rating work, a special engine is employed, and the test conditions are closely specified. The test engine is a modification of that used for octane rating, the principal difference being the use of a different cylinder having a precombustion chamber the volume of which is varied to change the compression ratio.

It was mentioned above that the ignition quality of a diesel fuel has little influence on engine-output limitations. That is, fuel cetane number does not bear the same direct relation to the allowable output of diesel engines that the fuel octane number bears in spark-ignition engines. The principal benefits of a high cetane number are better

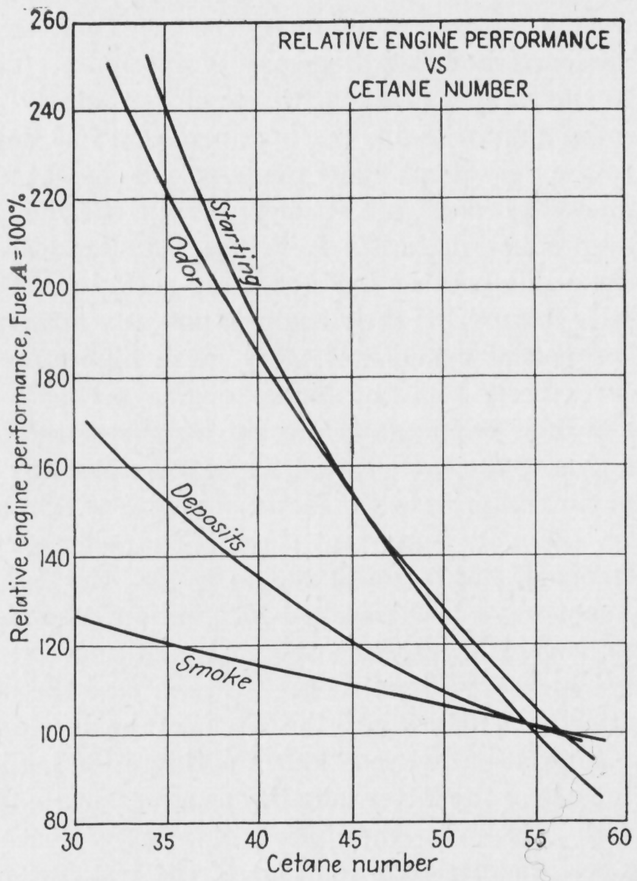


FIG. 137. Effect of cetane number on various engine-performance factors. (*Bogen and Wilson, Petroleum Refiner, Vol.* 23, 1944.)

starting characteristics and improved combustion, giving reductions in smoke, odor, and combustion-chamber deposits. Average values for these effects are shown in Fig. 137 for representative automotive engines.[15] Differences in injection-nozzle and combustion-chamber characteristics may cause individual engines to be affected to a greater or lesser degree than indicated, but Fig. 137 does give a good qualitative picture of the situation.

The ignitability of a fuel may be improved by the addition of any of quite a number of ignition accelerators. The average improvement in cetane number obtainable from three of these compounds is shown in Fig. 138. Note that all three compounds are oxidizing agents so that their effectiveness is not surprising.

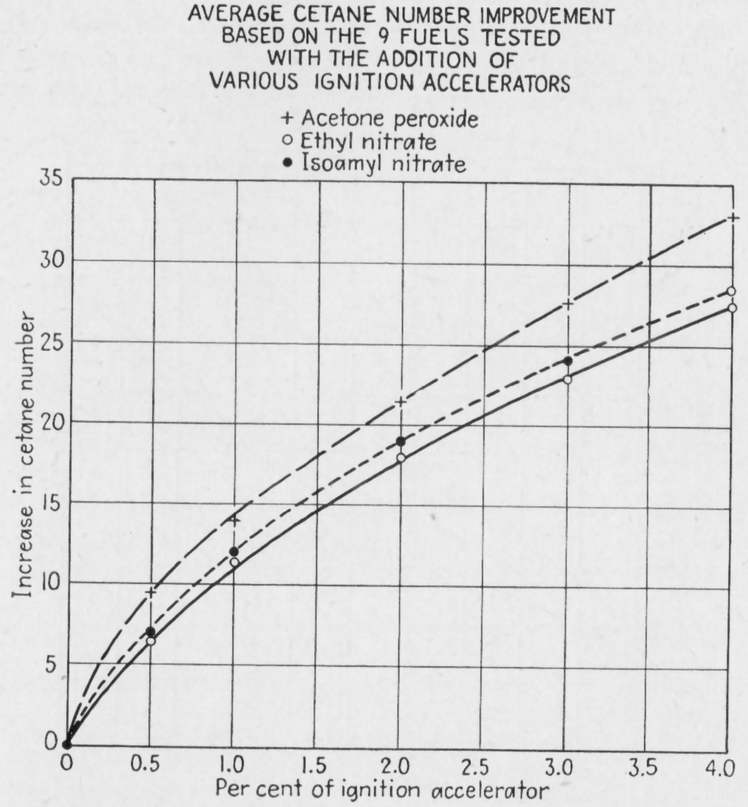


FIG. 138. Average improvement in the cetane number of fuels produced by three typical ignition accelerators. (*Bogen and Wilson, Petroleum Refiner, Vol.* 23, 1944.)

FUEL SPECIFICATIONS AND THEIR SIGNIFICANCE[16]

Volatility. The importance of fuel volatility has been referred to many times in previous pages. Since automotive fuels consist of mixtures of compounds (including some that would be gases at room temperature if they were not dissolved in heavier hydrocarbon liquids), no single scale of values can describe the volatility of any common fuel. It will not have a definite boiling point, for example. Instead, if a sample is placed in a flask and distilled, the temperature of the liquid and vapor in the flask will be found to increase with the percentage distilled off to the condenser. The standard ASTM test calls for measurement of the temperature as a function of the amount of condensate collected from a 100-cc sample, with the boiling rate adjusted to give 10 cc recovered every 1½ to 2 min. The *initial*

boiling point is taken as the vapor temperature as read in the flask at the instant when the first drop of condensate falls into the graduate under the condenser. The *end point* is the highest temperature indicated during the test. The apparent temperature of the vapor in the flask falls after the last portion to be distilled off leaves the flask during the latter part of the test. This is because the primary source of heat flowing into the thermometer is from the condensation of

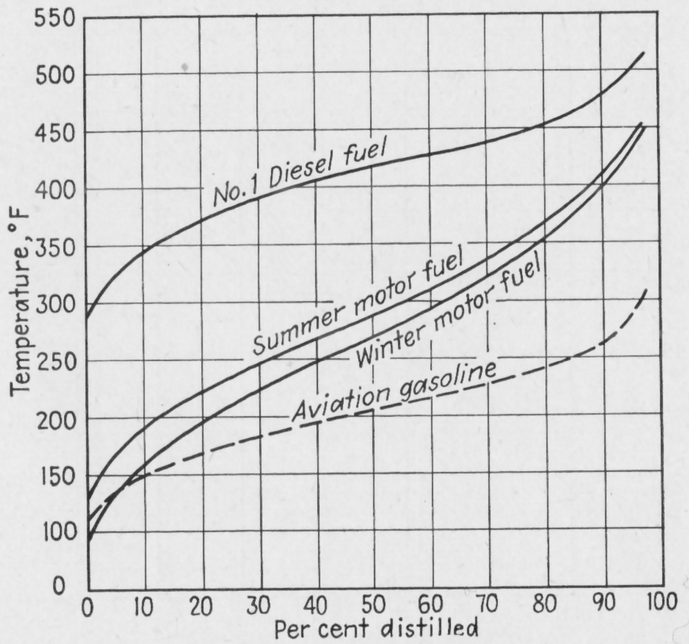


FIG. 139. ASTM distillation curves for a good diesel fuel, two automobile gasolines (motor fuels), and an aviation gasoline.

vapor on the thermometer bulb; when vapor ceases to condense on it, its temperature drops. The portion of the sample that remains in the flask is called the *residue*. Seldom greater than 1.5 per cent, it consists largely of very high boiling point hydrocarbons.

Distillation curves for a summer gasoline, a winter gasoline, an aviation gasoline, and a high-grade diesel fuel oil are shown in Fig. 139. The range of volatility permitted by the shape of the curves shown is desirable from the standpoints of both the refiner and the engine user. The former is able to use a fairly wide "cut" from his fractionating towers for any particular type of fuel. The latter receives a fuel having good cold-starting characteristics, coupled with minimum tendencies toward evaporation losses. In practice, it has been found that economic considerations make it worth while for refin-

ers to produce fuels having rather high end points, considerably higher than would be considered optimum were it not for the matter of cost.

The ASTM distillation curve described above is easily run and gives a good indication of the volatility characteristics of the fuel. Since in the engine the fuel evaporates in an atmosphere consisting primarily of air, a better picture of the fuel volatility is given by heating it in the presence of air so that the amount of fuel tending to

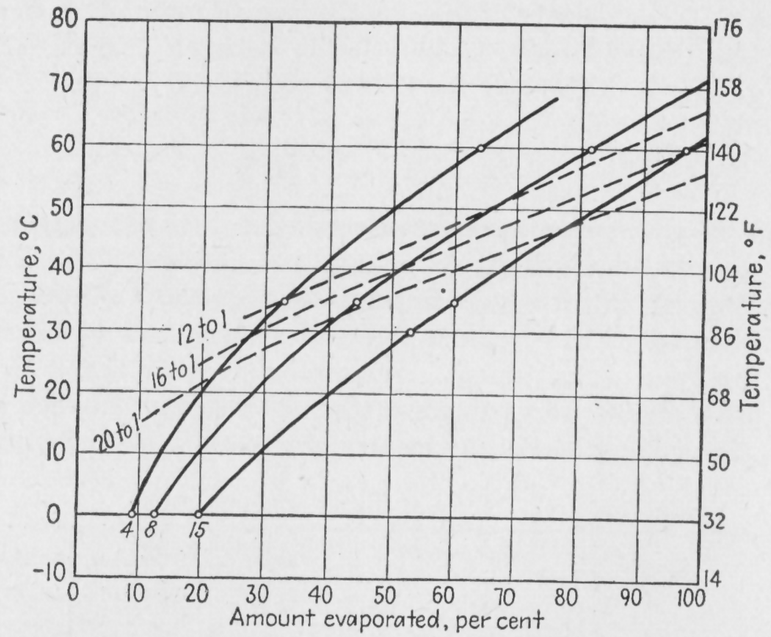


FIG. 140. Equilibrium–air-distillation curves for an automotive gasoline. The dotted curves give the fuel-air ratio of the fuel vapor-air mixture at any given temperature and percentage of fuel evaporated. The solid curves give the ratio of the weight of air to the total weight of both the vapor and the liquid components of the fuel. (*Bridgeman, Trans. SAE, Vol.* 23, 1928.)

vaporize is a function of the partial pressure of the fuel vapor. A test procedure of this sort called the *equilibrium-air distillation test* has been worked out. It involves the determination of the percentage of fuel evaporated in the air-fuel mixture leaving the apparatus at each of a series of temperatures for a given weight-flow ratio for the stream of fresh liquid fuel and air admitted to the apparatus. The test can then be repeated on additional samples at each of a series of air-fuel ratios. The results of such a test on a gasoline similar to the summer motor fuel of Fig. 139 are shown in Fig. 140. Note the much lower temperature required for evaporation in the latter case. Since the ASTM distillation curve is much more easily run than the equilibrium-air distillation test and since the relations between the two have been well established, the ASTM distillation test is used for most purposes.

A quantity called the *Reid vapor pressure* is usually included in specifications, for it gives a good indication of the fuel's starting characteristics and its tendency to give high evaporation losses. The Reid vapor pressure is essentially the increase in pressure within a bomb containing a fuel sample and air after the bomb is sealed at 40°F and heated to 100°F. Typical values called for by specifications are given in Table 17.

Density. The density of fuels or oils is most often given in *degrees API* in the United States. The relation between degrees API and specific gravity is defined by the following formula:

$$°\text{API} = \frac{141.5}{\text{specific gravity at } 60°\text{F}} - 131.5 \qquad (11)$$

Table IV in the Appendix gives the relation between degrees API, specific gravity, and weight in pounds per gallon. The API scale may appear odd, but it is used because the density of a liquid is most easily measured with a hydrometer, and such a scale is the natural one for this type of instrument. A simple derivation based on Archimedes' principle will serve to show that the relation between specific gravity and a linear scale on the straight section of the hydrometer

TABLE 17. TYPICAL FUEL SPECIFICATIONS

Type of fuel	Motor gasoline	Aviation gasoline	Automotive diesel fuel
Octane rating, min	75	100	
Cetane rating, min	.....	.....	50
Flash point, °F, min	.....	.....	133
Tetraethyl lead, ml/gal, max	1.0	4.0	
Corrosion, copper dish	None	None	
Gum, copper dish, mg/100 ml, max	5	5	
Sulphur, per cent, max	0.10	0.05	0.10
Distillation temperature, °F			
10 per cent evaporation, min	149	.....	400
10 per cent evaporation, max	.....	158	
50 per cent evaporation, min	257	.....	510
50 per cent evaporation, max	.....	212	
90 per cent evaporation, min	392	.....	620
90 per cent evaporation, max	.....	257	
End point, max	.....	311	
Residue, per cent, max	2	1.5	
Reid vapor pressure at 100°F, psi max	10	7.0	
Freezing point, °F, max	.....	−76	
Pour point, °F, max	.....	.....	−30

which protrudes above the surface of the liquid is in the form of Eq. (11). The constants 141.5 and 131.5 are functions of the geometry and weight of the hydrometer.

Gum Content. Fuel specifications normally include one or more tests to make certain that the fuel has been properly treated in the latter stages of refining to prevent gum formation. Such tests may involve evaporation from an open dish to determine the presence of gum in the residue, or they may involve accelerated oxidation tests (often carried out by bubbling oxygen through a heated bath of the test fuel) to determine the gum-forming tendencies.

Sulphur. Sulphur may appear in a fuel because of its occurrence in the original crude. It may be present as elemental sulphur, in which case it is very corrosive. It is most often present in the form of mercaptans, which are malodorous organic sulphur compounds. The latter may be readily detected by the so-called "doctor" test, while corrosive elemental sulphur or sulphur compounds may be detected by looking for discoloration of a polished copper dish used to evaporate a sample.

Other Fuel Tests. Many other tests may be required in specifications. These include tests for heating value, freezing point, color, corrosion, tetraethyl lead content, and acidity.

REFINING OF LUBRICATING OILS

While the refining of lubricating oil once involved no more than a few finishing operations carried out on selected components from the material left over from the refining of gasoline, modern processes have become so specialized that some concerns have set up completely separate refineries for lubricating oil.

The nature of the refining operations required to turn out a good engine lubricant depends to a great extent on the character of the crude from which the oil is to be made. It happens that Pennsylvania crudes have exceptionally small percentages of sulphur compounds, asphaltenes, and resins—all of which are substances that are very undesirable in an engine oil. Refining of such crudes is thus a relatively simple problem. Mid-continent and West Coast crudes on the other hand require more complex processing because of their higher asphaltene, resin, and sulphur content. However, certain of these latter crudes have special characteristics that make them exceptionally good lubricants once the asphaltenes and resins are removed. Crudes from some wells, for example, yield detergent oils (oils that tend to carry decomposition products in suspension instead of deposit-

ing them on engine parts in the form of varnish and/or sludge). The best engine lubricating oils are usually blended from a number of carefully selected crudes.

Vacuum Distillation. All but the light lubricating oils must come from the residuum. One means of extracting the light and medium lubricating oils from the residuum without raising their temperature to the point where the desirable naturally occurring molecular structures will break down is to carry out the distillation under low pressures—sometimes as low as a fraction of an inch of mercury absolute. This is called *vacuum distillation*, a process widely used in lubricating-oil production.

Solvent Extraction. Oils having an SAE viscosity of about 50 or higher cannot be extracted from the residuum even by vacuum distillation without damaging their properties as engine lubricants. They can be removed, however, by selective solvents, after which the solvent can be distilled off to leave the lubricating oil. Solvents can also be used for separating the desirable lubricating-oil hydrocarbons from asphaltenes, resins, and waxes.

Solvent extraction depends on the fact that one of the distinguishing features of any one of the various types of hydrocarbons such as paraffins or naphthenes is its relative solubility in phenol, benzol, propane, or other liquids. Mixtures of these or other solvents are especially effective in removing certain classes of hydrocarbons from petroleum oils in refining work. Paraffins are relatively much more soluble in a benzene-sulphur dioxide solution than naphthenes and asphaltenes, for example; hence, a benzene-sulphur dioxide solution may be employed to remove naphthenes and asphaltenes from a lubricating-oil stock. Other solvents such as phenol and furfural are much more commonly used commercially.[17] Liquefied propane is used to dewax an oil and thus lower its pour point. Not only may selective solvents help improve the quality of the lubricating oil from a predominantly paraffinic crude, but they make possible the production of good lubricating oils from some crudes containing large percentages of naphthenes and/or aromatics.

EVALUATION OF ENGINE LUBRICATING OILS

Significant Properties. The three most important sets of characteristics of an oil from the standpoint of engine service performance are those relating to its viscosity, its "wetting" abilities, and its tendency to deteriorate. It will be shown in Chap. X that the oil affects the load-carrying capacity of an ideal bearing only through its viscosity. While the heat conductivity and specific heat of the oil do have minor indirect effects, these properties do not vary significantly for commercial lubricants. The wetting characteristics of an oil determine how well the oil will stick to a surface. The more tenaciously a film of oil will adhere to a cylinder wall or piston ring, for example, the lower the wear rate on those parts.[18] The oxidation resistance of an oil determines the rate at which it will deteriorate. Most oils tend to oxidize and form acids, tarry sludges, or resins.

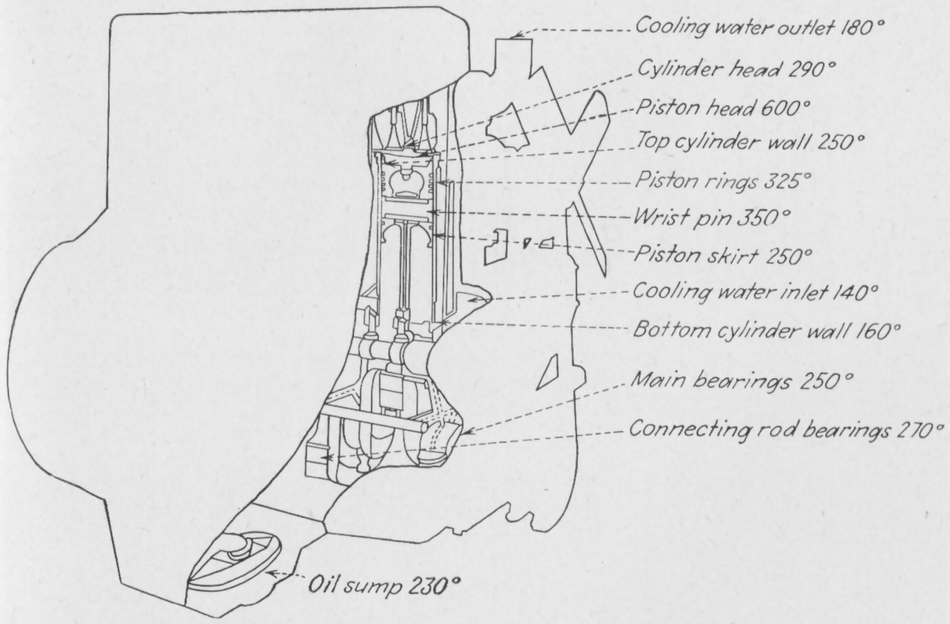


FIG. 141. Representative temperatures in deg F of the more important surfaces requiring lubrication in a typical engine. (*The Texas Co.*)

Acids tend to corrode engine parts, particularly some bearing metals, while the sludges and resins form coatings and deposits that tend to clog oil passages, cause piston rings to stick in their grooves, valves to stick in the valve guides, etc.[19] Sludges form flocculent tarry deposits, while the resins may deposit with the sludge or may form thin adherent coatings on engine parts. In the latter case the coatings are called *varnish* or *lacquer* because of their appearance.

The viscous properties of an oil are relatively stable and can be determined easily by standard tests. The wetting characteristics and oxidation tendencies, on the other hand, not only are difficult to measure because they are affected by so many engine-operating variables but are inclined to vary with the service history of the oil. Experience has shown that the only satisfactory tests for these properties are those run in engines under closely simulated or actual serv-

ice conditions. While useful chemical laboratory tests have been devised to aid in the selection of an oil, as a rule they are not adequate to distinguish between a good and a mediocre engine lubricant.

Engine-oil-performance Tests. In attempting to evaluate the performance of an oil, accelerated service tests may be run in an engine. The usual procedure is to run the engine for a standarized length of time under specified conditions and then disassemble it for

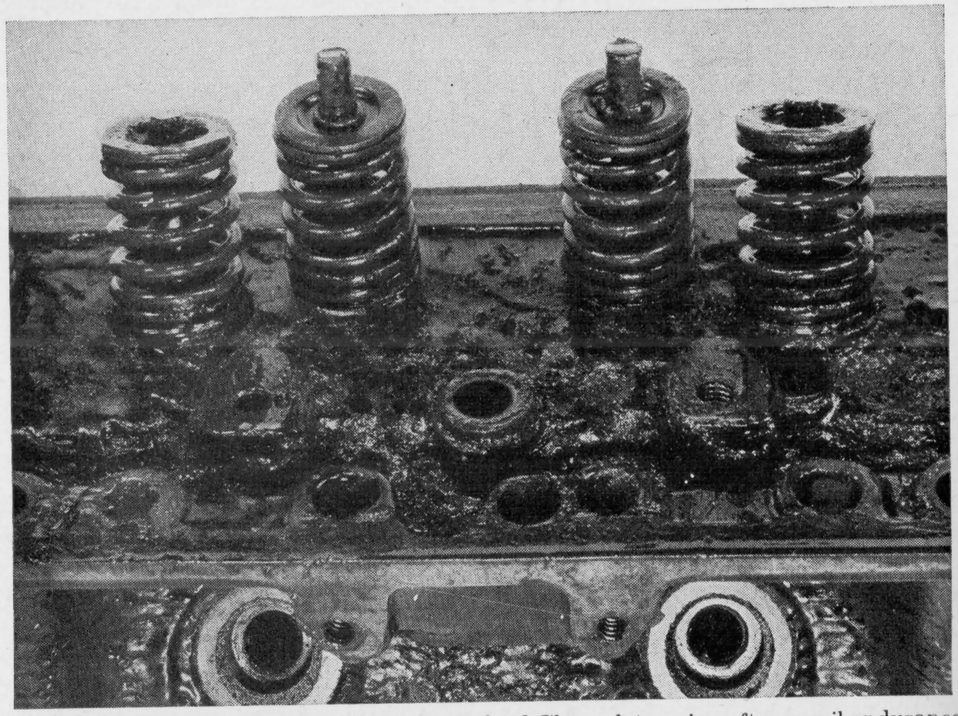
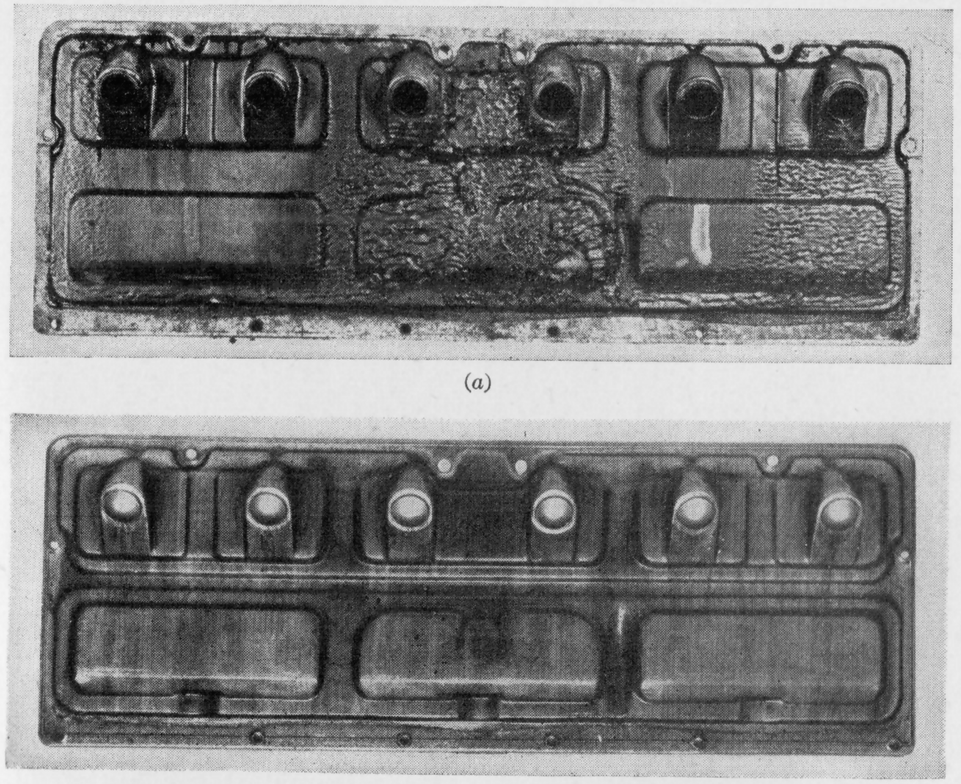


FIG. 142. Dirty valve gear in a heavily sludged Chevrolet engine after an oil endurance test. (*The Lubrizol Corp.*)

inspection. High outputs under high-temperature conditions usually cause the most rapid oil deterioration. In fact, oil temperature has such an important effect that an increase in crankcase oil temperature of only 20°F may double the rate of oil deterioration. On the other hand, too low a set of temperature conditions under idling or low-output conditions may also cause sludge to accumulate rapidly. High gas-blow-by rates past the piston rings into the crankcase under idling conditions are one cause of this condition. Another factor affecting oil deterioration is the temperature distribution in the engine, a variable not easily controlled in service tests. It is inevitable that some parts run much hotter than others—as shown in Fig. 141, the pistons, rings, and valve stems impose very severe conditions on the lubricant.

The critical parts such as these are the ones that must be inspected most carefully after an oil test.

Oil performance tests are often run in the form of cycles, the operating conditions being varied in sequence for specified periods throughout the test. Special cold sludging tests are usually run separately because hot operation might tend to clear the sludge considerably. After a prescribed endurance test the engine may be torn down and



(*a*)

(*b*)

FIG. 143. Effect of an additive on the amount of sludge deposited on the side plates of a Chevrolet engine during a standard oil endurance test. (*a*) Without an additive in the oil; (*b*) with an additive in the oil. (*The Lubrizol Corp.*)

inspected. The extent and nature of sludge and varnish deposits may be noted, and photographs such as those in Figs. 142 to 147 may be taken. The amount of wear may be determined by visual inspection or by dimensional inspection with a micrometer. An indication of the amount of corrosion of connecting rod and main-bearing inserts may be obtained by weighing the parts before and after the test.

The amount of engine operating time (usually 20 to 150 hr) coupled with the complete overhaul and the part replacements required make such a test quite expensive. Even so, one test is generally not

enough to evaluate the service performance of an oil—different engine makes and models may yield widely different results. A change in operating conditions in one engine may also have a much different or greater effect than in another. Thus it is evident that truly significant results can be obtained only after a long period of testing in a good laboratory operated by able and experienced personnel.

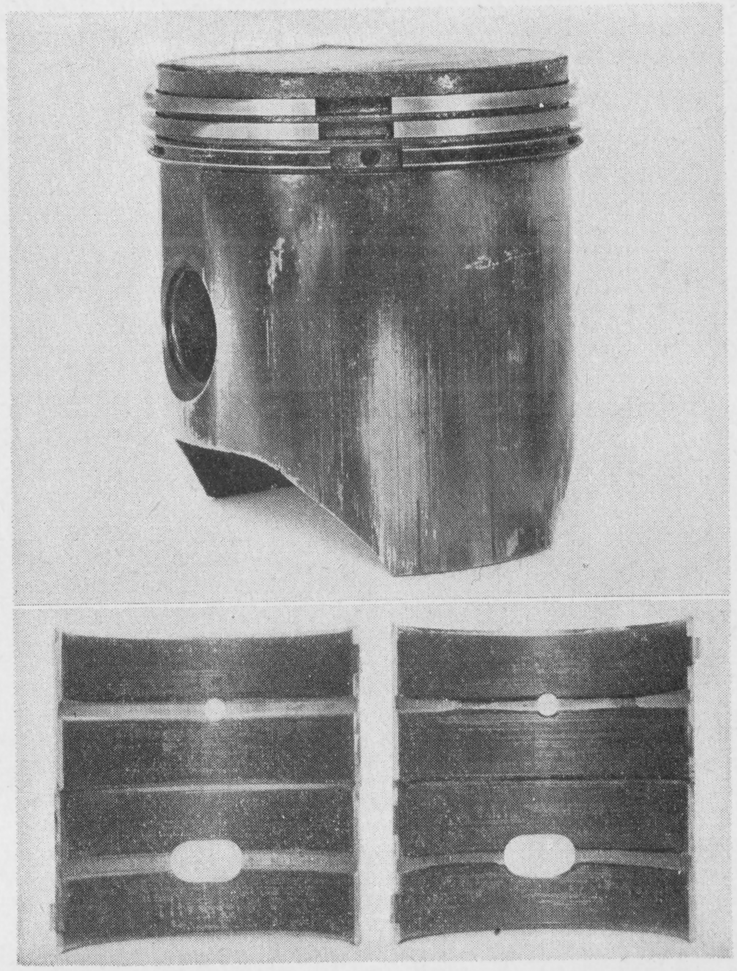


FIG. 144. Heavily lacquered piston and corroded main bearing from a standard type of oil endurance test in a Chevrolet engine. (*The Lubrizol Corp.*)

Condition of Engine Parts after Oil Performance Tests. The nature of the sludges and varnishes formed in engines is shown in Figs. 142 to 146. Figure 142 shows heavy sludge deposits on the valve gear of an overhead-valve engine after an oil-endurance test. Figure 143 shows the side plates that covered the push-rod galleries of two engines subjected to the same schedule of endurance testing.

Heavy sludge deposits were left by the test run with the unsatisfactory oil, while virtually none were left by the test run with the satisfactory oil. Comparative pictures of both bearing corrosion and lacquer deposits on pistons are shown in Figs. 144 and 145 after simi-

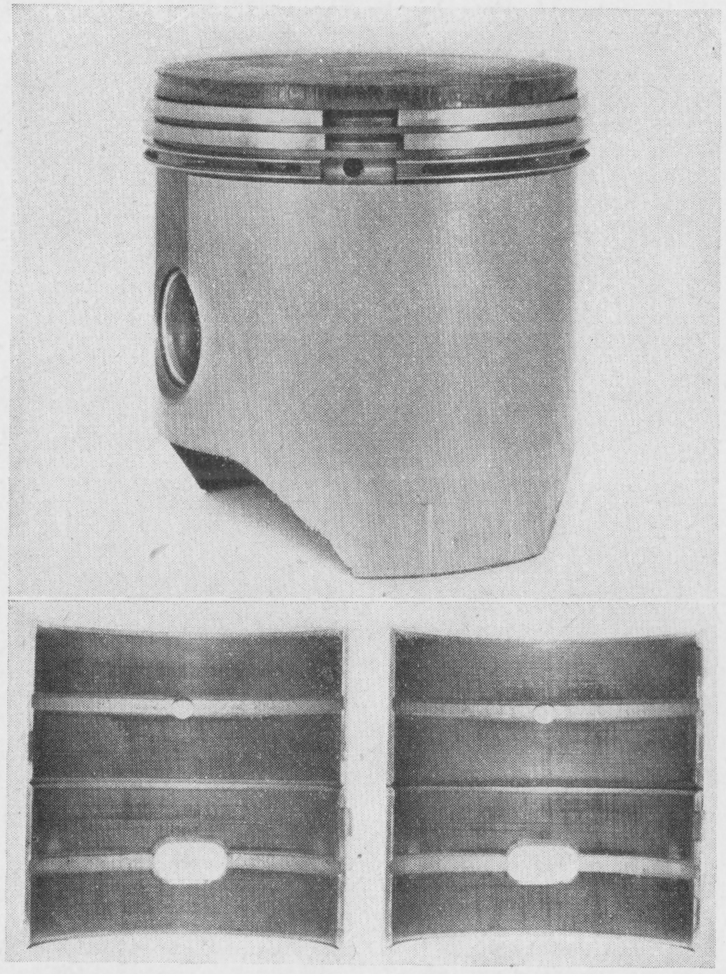
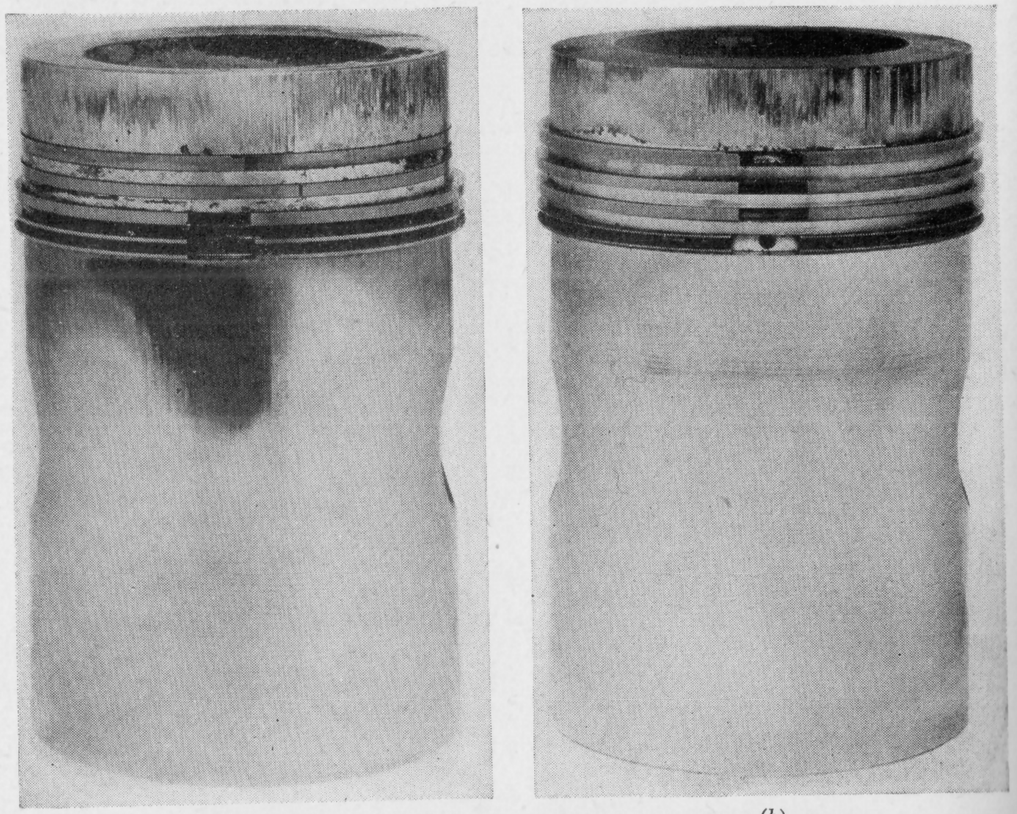


FIG. 145. A piston and main bearing from the same type of test as the parts in Fig. 144 but with an additive in the oil. Note that the piston skirt is free of varnish and the bearings show no signs of corrosion. (*The Lubrizol Corp.*)

lar tests. The better oil left almost no lacquer and caused no corrosion. The unsatisfactory oil left a heavy lacquer deposit on the piston skirt and severely corroded the bearing linings. Resinous deposits such as those on the piston skirt in Fig. 144 are especially likely to occur on the piston rings and in the ring grooves. Such deposits are likely to cause the rings to stick in their grooves and thus ruin their effectiveness as a gas seal. Figure 146 shows two pistons from a single-cylinder Caterpillar diesel engine of the type widely used in oil-

endurance-test work. After similar periods of operation the piston run with the satisfactory oil showed scarcely any lacquer deposit on the skirt, and the rings were quite free. The other piston run with the unsatisfactory oil showed heavy lacquer deposits on the skirt, and such heavy deposits in the ring grooves that the upper rings were



(*a*) (*b*)

FIG. 146. Typical pistons from a single-cylinder diesel test engine after similar endurance tests with different oils. Varnish and sludge deposits were so heavy on the rings and in the ring grooves of the piston in (*a*) that the upper two rings were tightly stuck in their grooves and the lower rings were partially stuck. Note that the varnish on the ring lands was so brittle that much of it has flaked off. (*The Lubrizol Corp.*)

tightly stuck and the lower rings partially stuck in their grooves. Such a condition always causes a loss in power and high oil consumption. In spark-ignition engines it also causes oil fouling of the spark plugs.

The most sensitive indication of the tendency of the oil to adhere to engine parts and prevent lubrication failure is given by the piston rings.[18] Partial breakdown of the oil film will cause these parts to become "scuffed." In severe cases the piston rings, piston, and cylinder may be heavily scratched and scored, as was the case during

the test from which the parts at the left of Fig. 147 were taken. Scuffing and scoring may occur even with the best lubricants if service conditions are too severe. These conditions are discussed in greater detail in Chap. XVI along with other types of engine-part deterioration.

Additives for Engine Oils. It has been found that small amounts of certain chemical compounds have pronounced effects on the service

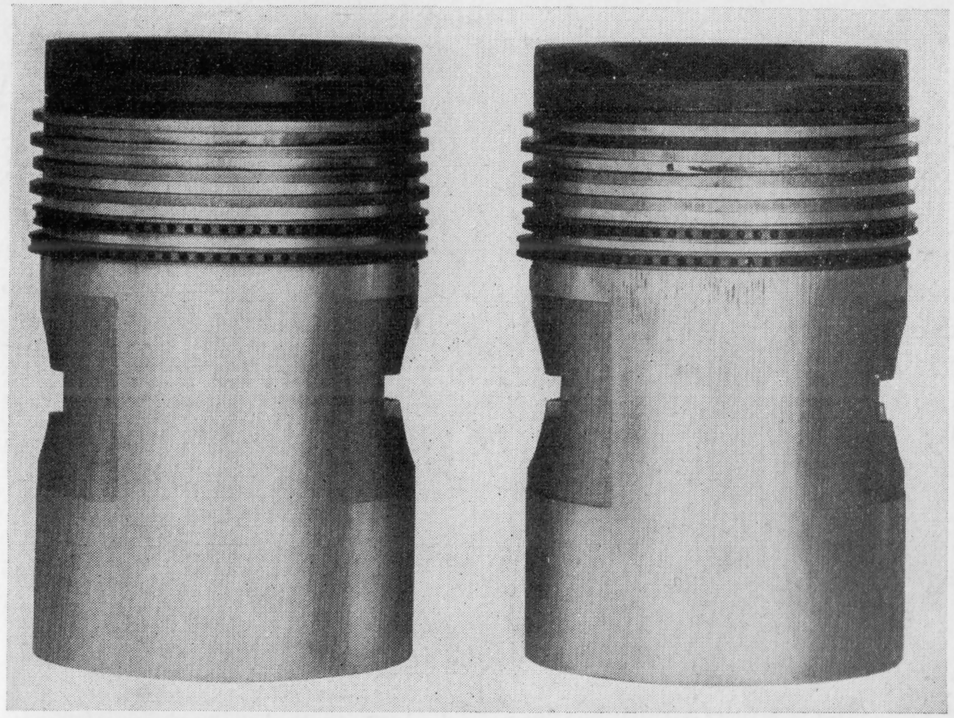


FIG. 147. Pistons and rings from the same engine as those of Fig. 146 after the same type of endurance test. The piston and rings at the left are in excellent condition, but both the piston and rings at the right have been scuffed. In the picture the vertical scratches from scuffing are clearest on the second ring and on the piston just below the ring belt. The scuffing resulted from the poor lubricating qualities of the oil. (*The Lubrizol Corp.*)

performance of an engine lubricant. It is largely because of this that some crudes make much better engine lubricants than others even though the refined products in both cases may have substantially the same proportions of paraffins, naphthenes, and aromatics. The best petroleum-base engine lubricants are made by blending oils from several carefully selected crudes and adding small amounts of particular compounds to give even better service-performance characteristics. Oxidation resistance can be improved by the addition of certain chlorinated esters. Other compounds may be added so that sludge or resinous varnish if it forms on such high-temperature parts as piston

rings will be emulsified and carried in suspension. If an oil tends to do this, it is called a *detergent* oil. Thus one oil that appears much darker and dirtier than another oil after a given test may leave a much cleaner engine.

Additives may be used in an oil to improve its wetting characteristics. Animal and vegetable oils or closely related compounds such as stearic or oleic acid are sometimes used because the long molecules of these compounds are polarized, the ends containing an OH group being attracted electrostatically to metal surfaces. Phosphorus and sulphur compounds may be added in small amounts to form chemically complex films of microscopic thickness on the metal surfaces. These films have lubricating characteristics similar to those of graphite and hence provide some lubrication if the oil film normally present breaks down in a limited area. Thus trouble can be prevented in many cases in which momentary local overloading would otherwise cause a surface failure and scuffing or scoring of the parts. The effectiveness of additives in reducing varnish and sludge deposits, inhibiting corrosion, and increasing lubricating-film adherence is indicated by the comparative pictures in Figs. 143 to 147.

LUBRICANT SPECIFICATIONS AND THEIR SIGNIFICANCE[16]

Viscosity. The viscosity of the lubricant determines the load-carrying capacity of any given bearing. It happens that it is also

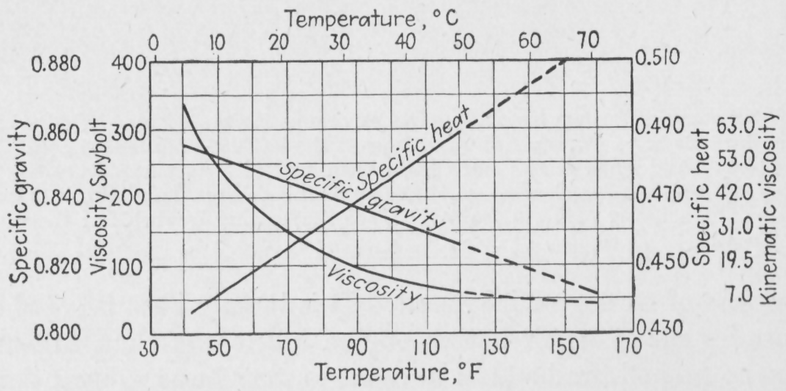


FIG. 148. Effect of temperature on the viscosity, specific heat, and specific gravity of a petroleum lubricating oil. (*Tichvinsky, Trans. ASME, Vol.* 60, 1938.)

closely related to low-temperature-starting troubles; the less viscous the lubricant, the more easily an engine can be started in cold weather. Viscosity may be expressed in poises (dyne-seconds per square centimeter), reyns (pound-seconds per square inches), or Saybolt Universal seconds (SUS). The first two are in absolute units, while the third is

simply the time required for a given quantity of oil to drain through an orifice in a Saybolt viscosimeter. The viscosity in Saybolt seconds is proportional to the *kinematic viscosity, i.e.*, the absolute viscosity divided by the density of the oil. Conversion factors for changing from one to another of these various units are given in Table II of the Appendix.

If the viscosity of an oil is determined at various temperatures, the data may be plotted on linear coordinates to give a curve such as that in Fig. 148. A more convenient set of special coordinates has been deduced theoretically to reduce the viscosity curves for all oils to what are very nearly straight lines. The viscosity data of Fig. 148 might be replotted on these coordinates to give a curve similar to one of those in Fig. 149.

Viscosity Index. The slope of the viscosity-temperature curve on the coordinates of Fig. 149 varies with the type of hydrocarbon, the paraffinic oils giving flatter slopes than oils having large percentages of naphthenes. Since engine temperatures are highest at high loads and high speeds, the oil thins out under just the conditions that a high viscosity is most desirable. It follows from this that the best lubricant is the one having the least change in viscosity with temperature. When it became evident that the slope of the viscosity curve should be specified, the slope of a set of parallel curves for a group of good paraffinic oils was arbitrarily assigned a value of 100, while that of a set for a group of highly naphthenic oils was assigned a value of zero. The *viscosity index* of any test oil was then defined as the relation its Saybolt viscosity bears to the viscosities of the corresponding arbitrarily defined paraffinic and naphthenic oils at 100°F if all three have the same viscosity at 210°F (see Fig. 149). Thus, the less the slope of the viscosity-temperature curve, the higher the viscosity index. This definition permits extrapolation of the scale to values above 100. Such oils are commercially available. In fact, some of the synthetic lubricants have exceptionally high viscosity indices, as may be seen in Fig. 149.

Pour Point. If an oil is gradually cooled, it will be found that at some definite temperature it will congeal and cease to pour. The temperature at which this occurs is called the *pour point.* It is a very important characteristic of a winter lubricant, for, if the oil once congeals, it becomes extremely difficult even to "turn the engine over" (*i.e.*, rotate the crankshaft), let alone start it.

The pour point of an oil can be reduced by diluting, chilling, and carefully filtering out the wax in the latter stages of the refining proc-

ess. It is also possible to put in one of several special chemical additives called *pour-point depressants*, which have proved to be very effective.

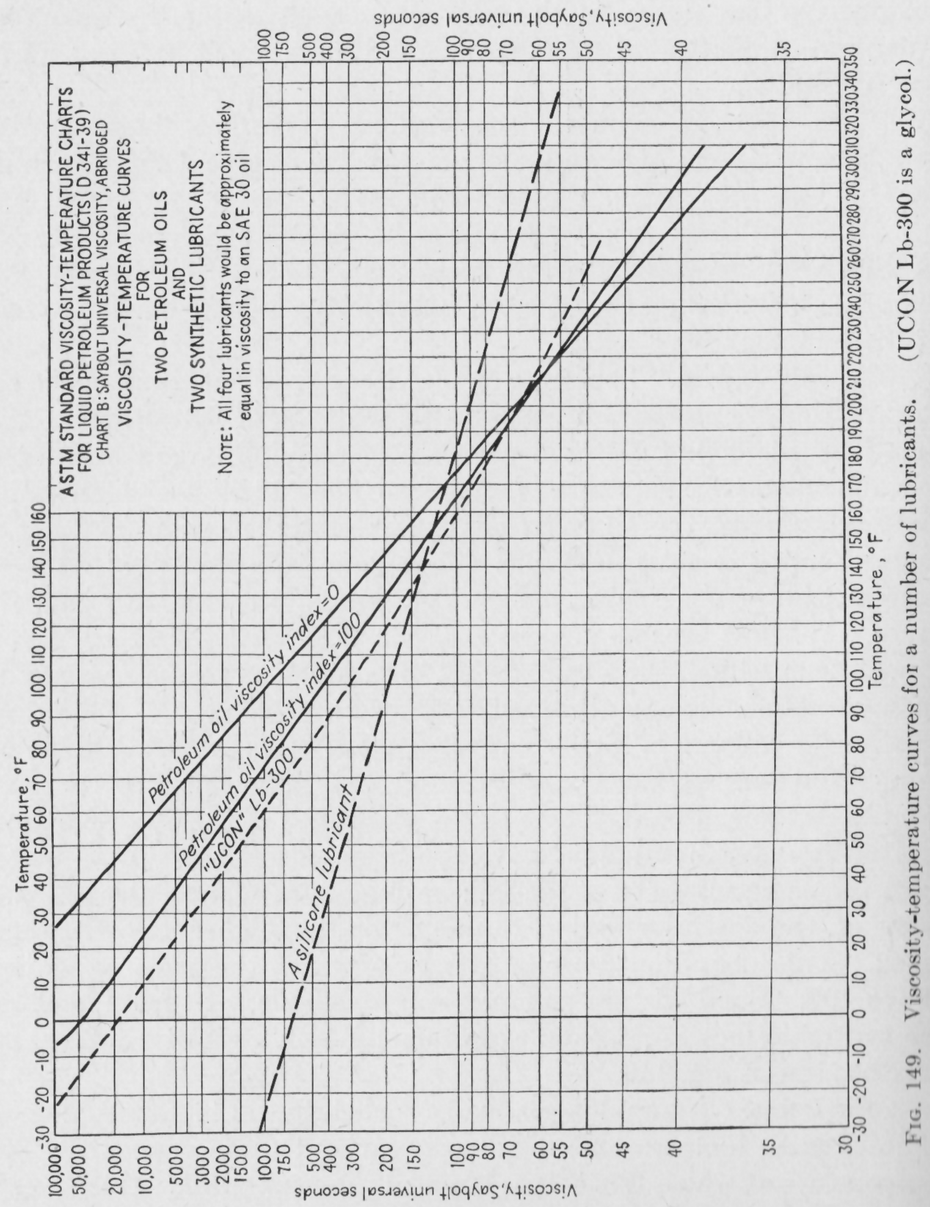


FIG. 149. Viscosity-temperature curves for a number of lubricants. (UCON Lb-300 is a glycol.)

SAE Viscosity. Commercial lubricants are classified as to viscosity according to a system adopted by the SAE. Any given lubricant is assigned a viscosity number according to the range in which its viscosity curve falls. This SAE viscosity number, or grade, is approximately equal to one-half the viscosity of the oil in Saybolt seconds at

210°F. Table 18 shows these SAE viscosity classifications. Two main groups of lubricants are provided for, one summer and one winter. Although the general viscosity characteristics of corresponding members of the two groups are the same, those in the latter must meet rather stringent pour-point requirements to ensure good cold-starting characteristics. These winter lubricants are designated by the letter "W," *e.g.*, "SAE 20 W" oil.

TABLE 18

SAE viscosity number	Viscosity range, SUS					
	At 0°F		At 130°F		At 210°F	
	Min	Max	Min	Max	Min	Max
10 W	5,000	10,000				
20 W	10,000	40,000				
10	.....	.....	90	Less than 120		
20	.....	.....	120	Less than 185		
30	.....	.....	185	Less than 255		
40	.....	.....	255	.....	...	Less than 80
50	.....	.....	...	.....	80	Less than 105
60	.....	.....	...	.....	105	Less than 125
70	.....	.....	...	.....	125	Less than 150

Flash and Fire Points. Early refining practices did not always remove the bulk of the highly volatile hydrocarbons from oils. Therefore, to avoid a possible fire hazard, oil specifications called for tests that would indicate the tendency of the oil to give off ignitable vapors. While there has been much less need for such tests in recent years, they are still called for in most specifications. The tests are carried out by heating a sample in an open dish. The flash point is the temperature of the oil at which it gives off enough vapor so that the vapor-air mixture above the oil will "flash," or burn momentarily, when a flame is introduced. The fire point is the temperature of the oil at which vapor is given off at a rate sufficient to maintain a continuous flame.

Sulphur. Even more than in gasoline, sulphur is very undesirable in a lubricating oil because it tends to corrode most bearing materials. Lubricating-oil specifications usually call for one or more different tests that are measures of sulphur content, including a test for the

percentage of total sulphur present and a test in which a brightly polished copper strip is heated in a sample of the oil. Darkening or pitting of the strip indicates the presence of corrosive material, *i.e.*, sulphur or sulphur compounds.

Neutralization Number. Acidic compounds may be present in new oils and are invariably present in used petroleum oils because of the tendency of many hydrocarbons to oxidize and form organic acids. Specifications usually required a determination of acidity on new oils, while a similar test on used oils has been found to give a good indication of the extent to which the oil has deteriorated in service. In both instances, an appreciable amount of acidity indicates the likelihood of bearing corrosion if the bearings are of the copper-lead or cadmium-silver variety. Brass, bronze, iron, and steel parts of an engine are unaffected.

The acidity of an oil may be given in terms of its neutralization number, *i.e.*, the weight in milligrams of potassium hydroxide required to neutralize 1 g of oil.

Carbon Residue. Some indication of the degree to which an oil will tend to leave carbon deposits in such critical spots as the piston-ring grooves is given by the carbon-residue test. The test procedure involves the heating of an oil sample in a system of crucibles specially arranged to minimize the amount of air to which the oil will be exposed. Heat is applied until all the oil has been either vaporized or broken down into carbon. The amount of the residue is expressed in grams per 100 g of sample.

Precipitation Number. Suspended solids or emulsified water may be separated from an oil by centrifuging, the amount precipitated out being expressed in milliliters per 100 ml of sample. Such a test may be applied to a new oil as a check on its cleanliness or to a used oil to determine its tendency to carry water or sludge in suspension. As is the case with most oil tests, the precipitation number of a used-oil sample means little by itself. An oil that leaves heavy sludge deposits in the engine may have a much lower precipitation number than, say, a detergent oil that would leave scarcely any engine deposits. The latter would probably be the better oil in spite of its "dirtier" appearance.

Oxidation Tests. Many types of accelerated oxidation test have been carried out in an effort to obtain a chemical laboratory method of evaluating the tendency of an oil to oxidize in an engine. The usual procedure is to bubble air or oxygen through heated samples of the oil. Correlation with engine service tests has been quite satisfactory for some specialized applications, but no one procedure has gained widespread acceptance. This is as might be expected, for engine operating conditions vary widely with the type and model of engine.

SPECIAL LUBRICANTS

Graphite. The flat, scaly crystals of graphite slip across one another with an exceptionally low coefficient of friction. Lightly loaded bearings operated at speeds sufficiently low so that the amount of heat generated is not a problem often may be lubricated satisfactorily with graphite alone. Finely divided graphite is used in some greases partly to improve their load-carrying characteristics and partly to leave a lubricating film of graphite on the parts. Some oils are available with colloidal graphite in suspension. One would expect the graphite in such lubricants to have little beneficial effect except where boundary lubrication conditions are a problem. Thus such parts as cylinder walls and piston rings, valve stems and guides, and speedometer cables would seem to offer likely applications. However, both piston rings and valve stems are in high-temperature regions, where excessive deposits of solid material left by evaporating lubricant will cause sticking and malfunctioning of the parts. This problem has presented a serious obstacle to the use of graphited oils in engines.

Greases. Where design considerations make oil-bath lubrication difficult or impractical and where the amount of lubricant required will not be too great, as in ball or roller bearings, it has been found that a grease will usually do very nicely. The grease will tend to remain in a semisolid condition until the parts begin to move and the frictional heat generated raises the temperature. This heat tends to be localized to the particular region in which the film of grease is being sheared, for as soon as the temperature in that film rises the viscosity drops very rapidly. It is also true that the viscosity of an element of grease is a function of the amount of "working," or shearing, that it has undergone in its immediate past. This factor tends to reduce further the viscosity of the grease on the rubbing surfaces and thus to localize the heating. Because of these characteristics, lubricant leakage through seals or glands is much less of a problem with grease than with oil unless excessive temperatures are encountered.

Greases are made by saponifying animal or vegetable oils with a base such as sodium hydroxide to form a soap and then mixing in two to ten times as much petroleum oil. The characteristics of the grease are dependent on those of the original petroleum oil, the original animal or vegetable oil, and especially the hydroxide used for the

saponification. Calcium-, sodium-, aluminum-, and lead-base greases are all fairly widely used. Each has its own particular characteristics, advantages, and disadvantages. Sodium-base greases are well suited to the higher temperature applications (above 175°F), for example, while calcium-base greases tend to be more water-repellent. Lead-base greases are good for applications where an extreme-pressure lubricant is required. Most greases contain more than one type of animal or vegetable oil and often more than one metallic base. The petroleum oil used is also usually a selected blend.

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Problems

1. Differentiate between distillation, fractionation, solvent extraction, thermal cracking, catalytic cracking, cyclization, and stabilization.

2. What significance does the term "77 octane" have as applied to a commercial gasoline?

3. Is it possible that one 80-octane fuel might be less resistant to detonation under some conditions than another fuel having an octane rating of 76 if tested in some particular commercial vehicle?

4. Does fuel volatility bear a direct and explicit relation to octane number if all types of gasoline are considered?

5. Distinguish between primary and secondary reference fuels.

6. List two advantages and two disadvantages to the use of (*a*) high-octane fuels; (*b*) high-cetane fuels.

7. In what way is it advantageous to have highly volatile hydrocarbons mixed in with less volatile hydrocarbons in a gasoline, (*a*) from the standpoint of the refiner; (*b*) from the standpoint of a motorist?

8. Using Dalton's law, explain the relation between the conventional ASTM and the equilibrium-air distillation curves.

9. From the standpoint of engine service performance, what is the significance of the ASTM distillation-curve 10 per cent point? the 50 per cent point? the 90 per cent point?

10. The float of a fuel hydrometer weighs 141.5 g. The volume of the bulbous lower portion is 179.7 cc, while the graduated stem has a cross-sectional area of 1.0 cm^2. If the stem is 13.8 cm long, determine the range of specific gravities that it could be used to measure, and the length of stem that would represent 1°API. Solve from the basic definition of density, *i.e.*, grams per cubic centimeter.

11. Indicate the significance of each of the following oil tests from the standpoint of oil service performance: viscosity, viscosity index, pour point, sulphur, and carbon residue.

12. Explain why endurance tests are run on engines to evaluate lubricating-oil performance. Indicate the oil properties that these tests bring out.

13. If a full-scale engine endurance test is run first with one oil and then with another and if the tests indicate that the second oil is superior to the first, is it possible that a different type of engine or a different set of test conditions might give just the opposite results? If so, indicate briefly why this should be the case.

CHAPTER IX

INTAKE AND EXHAUST MANIFOLDS

Intake and exhaust manifolds play an important part in determining engine performance, partly through their effects on volumetric efficiency and even more through their effects on the distribution of the quantities of both fuel and air to the various cylinders. To achieve an even distribution of both components of the charge is one of the most difficult problems in engine development work. As mentioned in Chap. VI, an even distribution of both fuel and air may be accomplished within ± 2 per cent if a fuel-injection system is used and if the intake and exhaust manifolds are carefully designed to give essentially the same gas flow passage resistance to and from all cylinders. Achieving a similarly good distribution in a carbureted gasoline engine is much more difficult.

In addition to the task of supplying the cylinders with uniform charges of fuel and air, induction systems must also take the air in quietly and clean it to remove abrasive dirt particles. After the working fluid has passed through the engine, the exhaust gases must be carried off and discharged quietly at a point from which they cannot poison the air for persons near the engine. The following sections cover the more elementary concepts involved in developing and working with equipment intended to carry out these functions.

FUEL VAPORIZATION

Vaporization Temperature Drop. The most important problems associated with spark-ignition-engine induction-system work are those of fuel vaporization. When a volatile liquid is sprayed into an air stream, the temperature of the air drops because heat is absorbed from the air as the liquid evaporates. Such a reduction in air temperature, of course, has an important effect on the evaporation process.[1] The magnitude of the air-temperature drop can be closely approximated by equating the product of the temperature drop, the specific heat, and the weight flow of the air to the product of the heat of vaporization of the fuel and the weight flow of the fuel. (This neglects the effects of any changes in temperature of the fuel between

the liquid and vapor states.) The relation may be expressed algebraically as

(Air-temp drop)(air-wt flow)(specific heat of air)
= (heat of vaporization of fuel)(fuel-wt flow)

Upon taking the specific heat of air as 0.24, substituting, and rearranging terms, the air-temperature drop then becomes

$$\text{Air-temp drop} = \frac{(\text{heat of vaporization of fuel})(F/A)}{0.24} \tag{12}$$

If octane is taken as representative of gasoline, the heat of vaporization is 131 Btu/lb (see Table 15). The expression thus becomes

$$\text{Air-temp drop} = \frac{131}{0.24}\left(\frac{F}{A}\right) = 546\left(\frac{F}{A}\right)$$

While most petroleum fuels give much the same vaporization-temperature drops as normal octane, the higher heats of vaporization

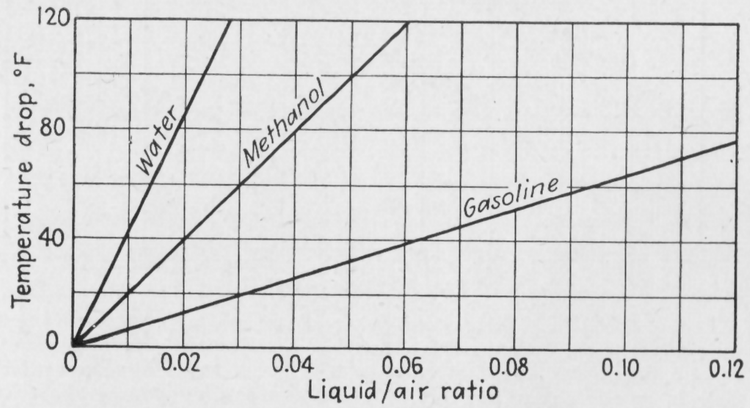

FIG. 150. Vaporization temperature drops resulting from the evaporation of a liquid injected into an air stream.

of such fuels as alcohol and benzol cause them to give considerably higher vaporization-temperature drops. The effects of other liquids are also of interest. Both water and water-alcohol mixtures have been injected into engine intake manifolds to inhibit detonation. Curves for the temperature drops accompanying the injection of a number of such liquids into an air stream are given in Fig. 150. Note the great effectiveness of water injection as a means of cooling an air stream. Charge cooling through liquid evaporation has been employed in many racing engines as a means of increasing air-charge density and hence engine power output. The cost of the high-volatile fuel required, however, renders this method of increasing engine output uneconomical for most purposes.

Dew Point of Fuel-Air Mixtures. The dew point of a fuel-air mixture is an excellent indication of the temperature at which relatively large percentages of fuel will pass through the manifold in the liquid state and hence is of paramount importance in mixture-distribution work. It may be recalled that the percentage of fuel to be expected in the liquid state at any temperature is shown in the equilibrium-air distillation curves of Fig. 140 for a representative gasoline.

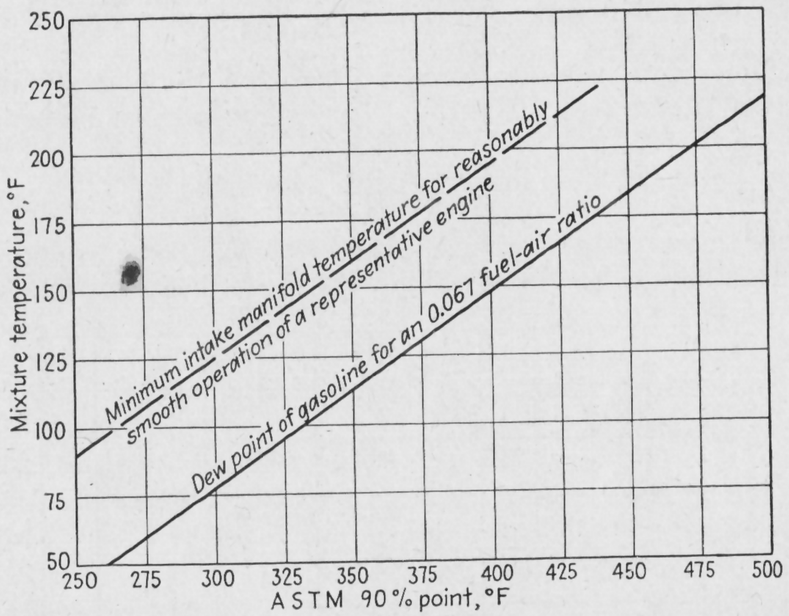

FIG. 151. Effect of the ASTM distillation curve 90 per cent point on both the dew point of the gasoline and the minimum intake-manifold temperature for reasonably smooth operation of one particular automotive engine. Under especially favorable circumstances in well-designed engines the minimum intake-manifold temperature for reasonably smooth operation can be dropped to a value as much as 40°F below the dew point curve shown here.

Since it takes much time to obtain such a set of curves even after the apparatus has been all set up, a project was carried out at the Bureau of Standards to relate the simple ASTM distillation curve to the equilibrium-air distillation curves.[2] It was found that the 90 per cent point of the ASTM curve was the most important factor. In fact, in spite of considerable variation in the shapes of the ASTM distillation curves for the various fuels tested in that investigation, very close correlation between the dew point of the fuel-air mixture and the 90 per cent point was always obtained. Figure 151 based on this work shows the relations between dew point, fuel-air ratio, and the 90 per cent point of the ASTM distillation curve. Since the usual run of automotive gasolines have 90 per cent points of the order of 320 to

400°F, appreciable percentages of liquid fuel in the manifold should be expected if the mixture temperature drops below about 150°F.

Wet-bulb Effects. Temperature measurement is always a difficult problem if the measurements must be made in a rapidly flowing mixture of air, vapor, and evaporating liquid droplets. The liquid droplets will impinge on and evaporate from any temperature-sensitive element placed in the stream, thus causing it to indicate the wet-bulb temperature of the mixture. The magnitude of this error, fortunately, is much lower with gasoline than with water because its heat of vaporization is only about one-sixth as great. While a detailed discussion of the problem is not in order here, it may be stated that the wet-bulb depression in engine intake manifolds seldom exceeds 15°F if the point of temperature measurement is removed a reasonable distance from the point at which the fuel is introduced. This permits the evaporation of the lighter ends of the fuel so that by the time the mixture reaches the sensitive element of the instrument the mixture temperature is not too far from the saturation temperature for the remaining liquid.

MIXTURE DISTRIBUTION IN GASOLINE ENGINES

Causes of Poor Distribution. Practically all the discussion in previous chapters has presupposed that all cylinders of multicylinder engines operated in the same way on the same fuel-air ratio and with the same amount of charge each cycle. This represents the ideal condition that might conceivably be approached. However, it is not difficult to imagine that fuel leaving the discharge nozzle of a carburetor may be deflected to one side by the throttle valve, that thus even before it leaves the carburetor the fuel-air mixture entering the intake manifold is stratified into rich and lean streams, and therefore that the mixture flowing to some cylinders may be much richer than that going to others. Such a condition is spoken of as *poor fuel distribution* or *poor distribution.* Another cause of segregation of the fuel and air may be found at the junction between the carburetor riser and the *runners, i.e.,* the horizontal passages in the manifold. In passing through this tee, the heavier fuel droplets tend to impinge on the manifold wall opposite the riser and flow along it in liquid form. The smaller fuel droplets may remain suspended in the air stream, but in passing around the bend they tend to be centrifuged to the outer portions of the air stream. *Because of these factors the mixture distribution in a carbureted engine is usually not uniform unless most of the fuel is evaporated before the mixture passes very far into the manifold.*

Since air velocities in the manifold run about 150 ft/sec, not much time is available for the evaporation process.

Manifold "Hot Spots." It has been found necessary in practically all carbureted multicylinder engines to provide some means of heating the mixture entering the manifold. This follows readily from the facts developed in the preceding sections. If a carburetor inlet air temperature of 70°F and a fuel-air ratio of 0.070 are assumed, these facts are as follows:

1. The fuel-vaporization-temperature drop will run about 40°F if all the fuel is to be vaporized.
2. The dew point of the vaporized mixture will run at a temperature of about 150°F.
3. The amount of heat required is equivalent to that needed to heat the air from 30 to between 100 and 150°F.

It happens that the exhaust manifold offers a convenient source of heat. The usual procedure is to provide a jacket around the junction between the riser to the carburetor and the intake manifold and allow exhaust gas to circulate through this jacket. A thermostatically operated valve is often employed to control the amount of exhaust gas circulated through the riser jacket and hence the heating effect. Further heating of the mixture may be obtained by placing the exhaust manifold directly below the intake manifold. Liquid fuel flowing along the bottom of the latter is vaporized by the heat radiated from the upper part of the exhaust manifold.

It is desirable to heat the mixture as little as possible, partly to give the maximum density of charge going to the cylinders and hence the maximum full-throttle power output and partly to inhibit detonation, for, the higher the mixture temperature, the lower the detonation limit. Because of this it is considered best to localize the heat to the junctions between the riser and the runners. Automotive-engine manifolds will handle an appreciable amount of liquid and still give fairly good mixture distribution if the hot spots are designed to give mixture temperatures of 80 to 110°F under peak power-output conditions. These low-mixture temperatures increase the full-throttle power output of an engine about 6 per cent as compared with the 150°F condition for which all the fuel would be evaporated. Increasing the output in this way requires little or no increase in production costs. At part-throttle conditions the mixture temperature will generally run higher with such an exhaust-jacketed manifold (see Fig. 83). This is desirable because the throttle can always be opened a little farther to offset the lower volumetric efficiency, and the higher temperature improves the mixture distribution.

Effect of Distribution on Engine Performance. To the engine operator, the most striking effects of poor mixture distribution are those evidenced by "rough" operation. Some cycles may deliver much less power than others or may even misfire. The irregularity of the power impulses induces torsional vibration of both the crankshaft and the engine. This vibration is transmitted through the engine mount or from the supports of the driven shaft and causes a pronounced increase in the vibration level in the vicinity of the engine.

Since other factors such as excessive spark gaps may display similar symptoms, in mixture-distribution-test work it is desirable to correlate observations on engine roughness with operating variables readily subject to instrumentation. The effect of distribution on the exhaust-gas composition is self-evident. A gas-sampling tap in each exhaust port can be connected to a selector valve and a sample taken for each cylinder in turn. The principal disadvantage of this procedure is the special apparatus and the time required for each set of observations. The temperature of some combustion-chamber element such as the spark-plug center electrode or a portion of the cylinder head also provides an indication. The very good correlation between the two procedures is shown in Fig. 152.[3] As may be suggested by Fig. 152, yet another method of detecting poor distribution is by examining the shape of the power curve of a mixture loop. Since the output of the engine as a whole is the summation of the outputs of the individual cylinders, the effect of poor distribution on the shape of such a curve can be visualized by plotting a set of curves of power vs. fuel-air for each cylinder of the multicylinder engine. Any given point on a curve for an individual cylinder will be displaced from its actual fuel-air ratio to the corresponding mean value for the engine as a whole. Inspection will show that the curve defined by the summation of the individual cylinder outputs under poor distribution conditions must have a more rounded peak with the maximum output at a higher fuel-air ratio than would be the case with perfect distribution. Figure 153 shows a pair of curves of this sort. Data are given for both very good and rather poor distribution conditions.

Nonuniform Distribution of Fuel Components. Distribution problems are often aggravated by the fact that the last portion of the fuel to evaporate consists of the "heavy ends," *i.e.*, the least volatile portion, of the fuel. The antiknock characteristics of these heavy ends are often markedly inferior to those of the more volatile fuel components. Thus the mixtures going to the various cylinders of the same engine may differ in antiknock value by as much as 10 octane numbers.

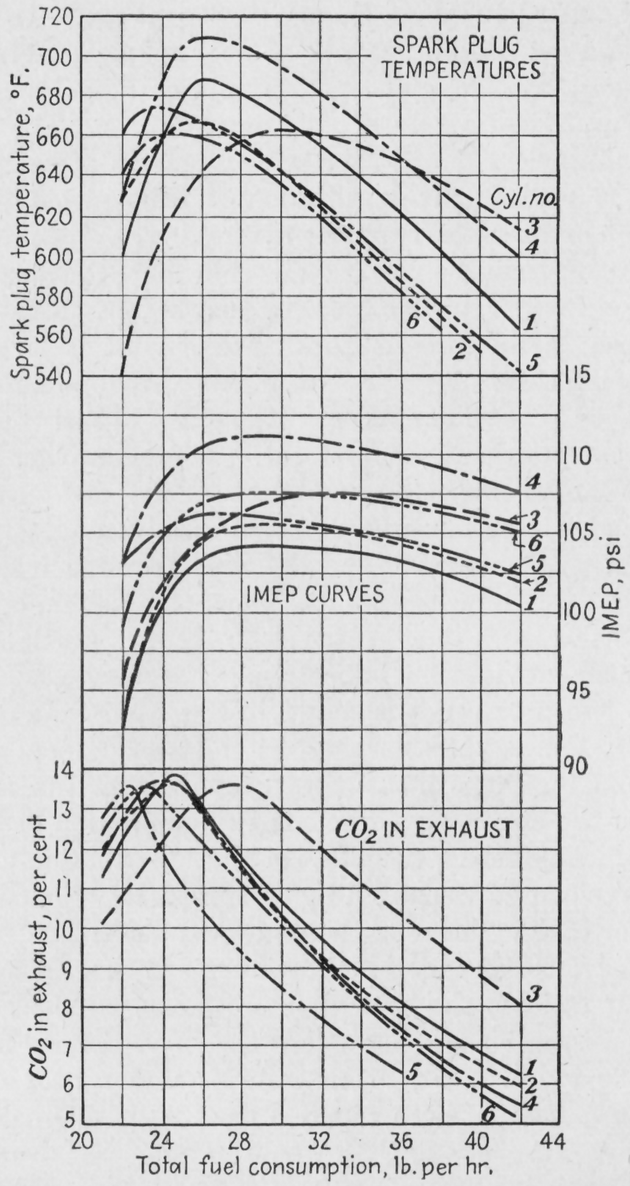

FIG. 152. For descriptive legend see opposite page.

Another form of poor fuel-component distribution may arise if the tetraethyl lead content of the fuel is high. Tetraethyl lead is normally added to a gasoline in the form of ethyl fluid, which contains ethylene dibromide in the chemically correct proportions to convert all the lead to lead bromide when the hydrogen and carbon in both compounds are burned during combustion. The lead bromide is quite volatile and passes off with the exhaust. If the tetraethyl lead

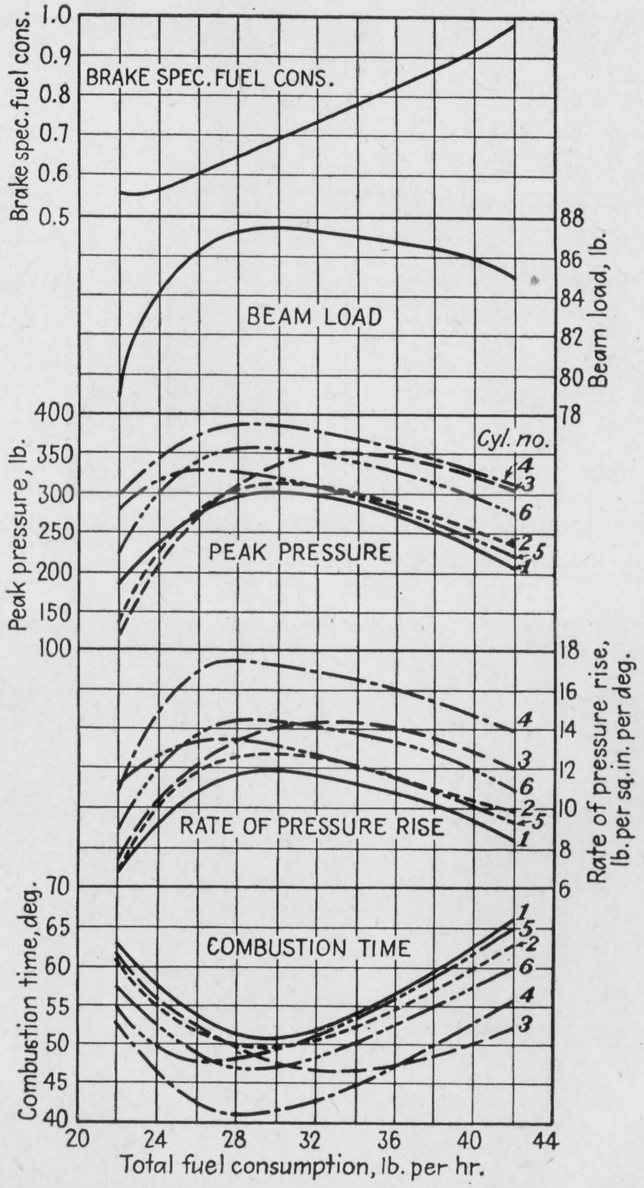

FIG. 152. Effect of fuel-flow rate to a spark-ignition automotive engine operated at constant throttle on performance parameters for each of the six cylinders. (*Rabazzana and Kalmar, Automotive Ind., Vol.* 66, 1932.)

were used alone, metallic lead would be left by combustion and would deposit on the combustion-chamber parts. Under such conditions the spark plugs may be affected very quickly by the formation of lead beads on the electrodes. These are likely to short the spark gap so that the plug will cease to fire. Such a condition is called *lead fouling* of the spark plugs. Figure 154 shows an example of this condition.

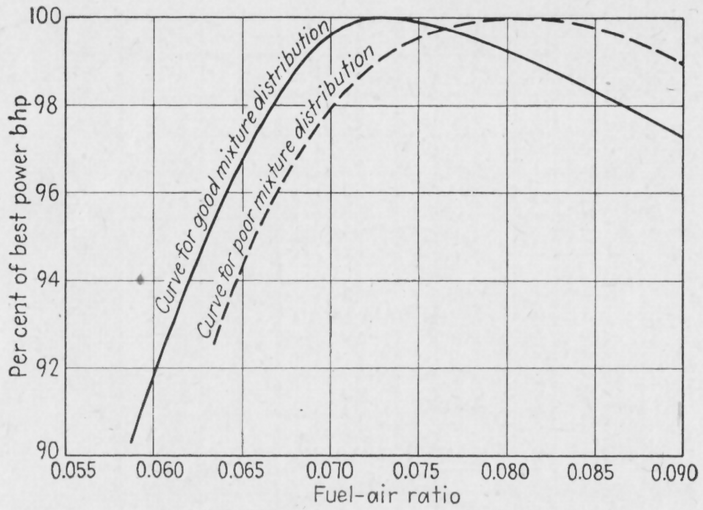

FIG. 153. Effect of mixture distribution on the shape of the power curve of a mixture loop.

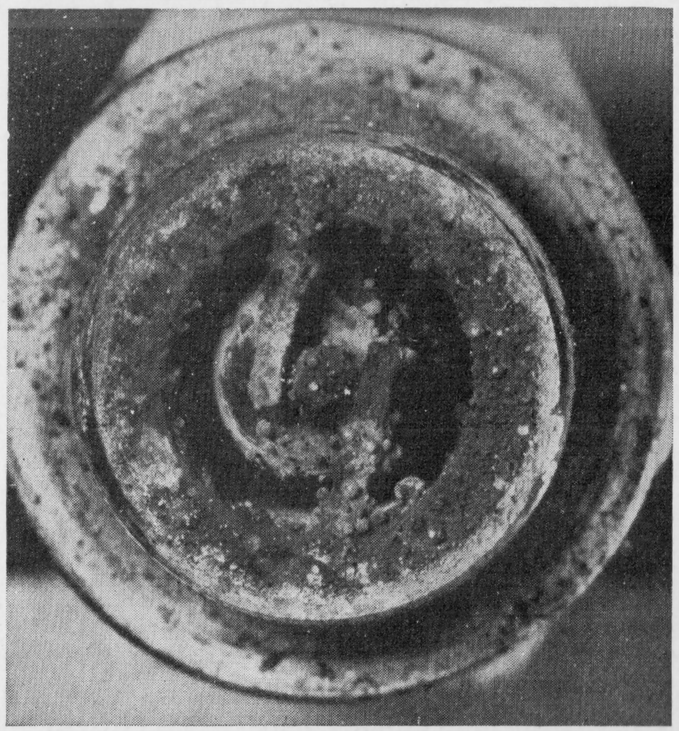

FIG. 154. Photograph of the electrodes of a spark plug badly fouled by lead.

Lead fouling of the spark plugs may be induced by poor distribution in the intake manifold, for the volatility of the ethylene dibromide is about the same as that of the lighter ends of the gasoline, while the volatility of the tetraethyl lead is about that of the heaviest fuel components. If most of the latter goes to one or two cylinders, there will not be nearly enough ethylene dibromide to combine with the lead to prevent lead fouling in that cylinder or cylinders. At the same time, other cylinders will be supplied with an excess of ethylene dibromide. The free bromine released in those cylinders will be very corrosive and will attack the exhaust valves. Both lead fouling and corrosion by ethylene dibromide have been very serious problems in aircraft engines in which the tetraethyl lead content of the fuel has run from 3.0 to 6.0 cc/gal.[4]

PROBLEMS IN INDUCTION-SYSTEM DEVELOPMENT

Since a more thorough treatment is not in order here, the following sections are intended merely to indicate the nature of the problems associated with induction-system development. These problems are very much interrelated—so much so that a change can seldom be made in one element of the system without affecting one or more of the other elements. This fact should be kept in mind in dealing with any specific problem such as one of those mentioned in the following sections.

Control of Liquid Flow through Manifolds. Many attempts have been made to handle wet mixtures in intake manifolds and still obtain good distribution. The results of this work have made it possible to handle moderate amounts of liquid and yet obtain tolerably good distribution.[3,5–7] *No arrangement, however, has been found satisfactory for mixture temperatures below about* 80°F *with commercial gasolines.* Below a temperature of this order the percentage of liquid fuel becomes so great and the vagaries of air-flow patterns with their swirls and crosscurrents become so unpredictable that a generally satisfactory solution short of fuel injection does not appear possible. Suppose, for example, that a particular set of manifold passages were developed and found to operate well with a very wet mixture under certain conditions. Then, if the attitude of the engine were changed from the horizontal to an angle of perhaps 15 deg to simulate a hill-climbing condition, the liquid-fuel-flow distribution would naturally be upset and excessive amounts of fuel would be supplied to the rear cylinders at the expense of those at the front.

While there are important limitations on what can be done in the

way of manifold design, it is still very true that, with any given fuel, some manifolds permit considerably lower mixture temperatures than others. One of the most effective devices is to get the fuel off the manifold walls and into the air stream in a finely divided state. One means of accomplishing this is the use of sharp corners instead of sweeping curves at all bends in the manifold, as indicated in Fig. 155. Fuel traveling along the wall tends to be swept off at these corners and mixes with the air stream. Another important rule is to avoid pockets or other regions in which liquid may collect. Backflow in the air stream builds up or permits the building up of a pool of fuel. The same thing might happen in the manifold of Fig. 155*a*, for the large variations in flow-passage area between the riser and the port would leave "dead spots," *i.e.*, regions of low velocity in which fuel would tend to collect. The usual procedure is to design the manifold to give a uniform air-stream velocity from the carburetor to the port of 200 to 300 ft/sec at the peak output condition for the engine. This will prevent the liquid droplets from coming out of suspension and hence the formation of liquid pools. Not only are these pools of fuel inclined to spill large droplets into certain particular cylinders, but they tend to do so irregularly, with many ill effects. A change in throttle opening, for example, will be inclined to change the air-flow pattern in the manifold and hence sweep the entire pool into one cylinder and thus cause misfiring. Note that the intake manifold on the engine of Fig. 10 is designed to drain liquid fuel directly to the cylinders to prevent the collection of liquid pools.

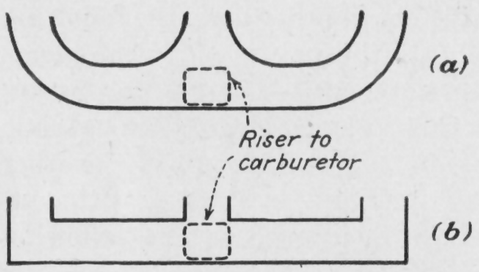

FIG. 155. An intake manifold with smooth sweeping bends (*a*), and one with sharp corners (*b*) designed to throw liquid fuel from the walls into the air stream.

The most effective means of obtaining good liquid distribution is to prevent the liquid and air streams from separating and taking different paths. This can be accomplished by joining the runners to the individual cylinders at a common point and maintaining a high velocity in that region. If the intake events of the cylinders fed from this common distributing center do not overlap appreciably, the mixture of air and liquid fuel will feed alternately to each of the cylinders in turn. High velocities at the distributing point coupled with a relatively intense hot spot at the junction will prevent the accumulation of liquid pools. Thus this arrangement will give good distribution of

the liquid droplets; for only one cylinder will be fed at a time, and all the mixture passing the junction between the runners and the riser will go to that cylinder. If the engine has more than four cylinders, the cylinders can be divided into groups, no more than four being supplied by any one riser and carburetor. Figure 156 shows a manifold of this type for an eight-cylinder V-engine. In this case a single carburetor having two venturis fed from a single float chamber has been found to be quite satisfactory.

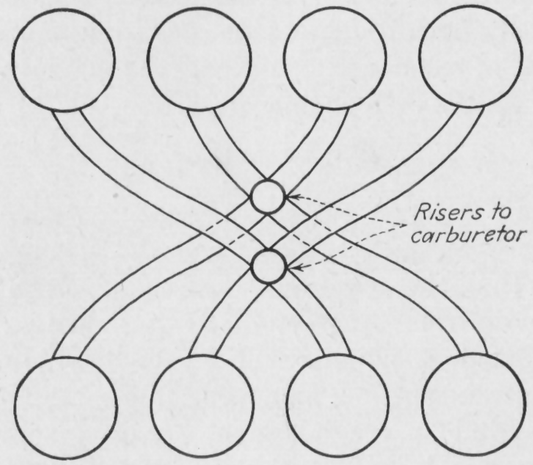

FIG. 156. Intake manifold for a V-8 engine designed to give good mixture distribution.

Resonance. Large pressure fluctuations may occur in intake manifolds as a result of air-column vibrations much like those in an organ pipe. These vibrations may aid or they may detract from the charging process, depending on whether the interior of the cylinder is in a high-pressure or a low-pressure field at the time when the intake valve closes.[8–11] Since the natural frequency of the induction system is fixed, the volumetric efficiency of the engine tends to fall off at speeds either above or below that at which resonance gives a "ramming" effect on the cylinder. For most engines the effect is small, or the peak effect is at a speed out of the operating range. Light-plane engines with their long intake pipes are an exception, however, and are usually designed to take full advantage of the possible improvements in volumetric efficiency. These special provisions for induction system resonance may yield an increase in bmep of 10 per cent or more.[12]

Air-charge Distribution. While most engine-distribution problems are those brought about by differences in the rate at which fuel is supplied to the various cylinders, it is also possible to have nonuni-

form air flow rates. Such a condition is usually caused by poor intake-manifold design. Two adjacent cylinders are often served by a common passage from the main portion of the manifold. This simplifies the castings and hence is desirable *unless* the charging processes for the two cylinders overlap. That is, if one of the two cylinders begins its intake stroke while the other is completing the same event, the former will rob the latter of its fair share of fresh charge and thus cause a substantial difference in output between the two cylinders even if the fuel-air ratios are the same. Such a condition will be characterized by both rough engine operation and a loss in power output. Because of resonance conditions, the magnitude of the effect is likely to vary widely with engine speed.

EXHAUST MANIFOLDS

Passage Size. Exhaust passages should be designed so that the pressure drop between each cylinder and the exhaust pipe is small. More important than the precise value of the pressure drop is its uniformity for all cylinders. Any condition that changes the amount of residual exhaust gas in a cylinder will change its volumetric efficiency and hence its power output. If an appreciable back-pressure variation exists between cylinders, rough engine operation is almost certain to result. Similarly, overlapping exhaust periods for adjacent cylinders discharging into a common passage are likely to leave a large amount of residual exhaust gas in the cylinder that is supposed to complete its exhaust event just as the other begins its exhaust blowdown.

Since the pressure drop through the exhaust system varies as the square of the flow rate, peak output at high speeds represents the critical condition. A small loss in power from exhaust back pressure is usually expected under these conditions. A back pressure of about 4 in. Hg is generally considered acceptable for a complete automotive exhaust system with the muffler.

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Problems

1. Consider an engine that is made up of four cylinders of the size and characteristics of the cylinder from which the data for Fig. 82 were obtained. Determine the effect of poor distribution on the mixture loop for such an engine if one of the four cylinders were to operate with a mixture 15 per cent leaner than the mean value for the engine, while the other three ran 5 per cent richer than the mean. Plot average imep vs. fuel-air ratio for both the given nonuniform-mixture-distribution condition and for perfect mixture distribution.

2. Estimate the temperature drop from vaporization of the fuel at a fuel-air ratio of 0.080 for (*a*) isooctane, (*b*) benzene, and (*c*) a special fuel consisting of 50 per cent benzene, 25 per cent methanol, and 25 per cent ethanol.

3. Estimate the minimum intake manifold temperature for complete vaporization of the fuel at a fuel-air ratio of 0.070 for a gasoline having the same volatility characteristics as that of Fig. 140.

4. Estimate the minimum intake manifold temperatures for complete vaporization of each of the two gasolines in Table 17, using Fig. 151. Assume a fuel-air ratio of 0.070 in each case.

CHAPTER X

LUBRICATION

An adequate and properly directed supply of lubricant is second in importance only to the supply of fuel and air in the operation of modern combustion engines. Even in the earliest automobile engines, careful attention had to be given to the lubrication problem—oil cups were provided on the main and connecting-rod bearings so that the driver could stop the engine every 25 or 50 miles and reach in through the sides of the open crankcases then in use to lubricate the bearings. When we consider how such a primitive lubrication system must have limited engine speed and output, we realize how important though inconspicuous a part modern lubrication systems play.

While, as implied above, the primary function of an engine-lubrication system is to prevent bearing failures even at high speeds and loads, it should also ensure low wear rates (*i.e.*, long service life), minimize power losses caused by friction, and serve as a means of cooling such parts as pistons and bearings. Thus in the larger sense the effectiveness of engine lubrication often limits allowable engine speeds and loads, largely determines service life, and plays an important part in determining power output and fuel economy. This chapter deals with these problems by first presenting the elementary concepts of lubrication and then by relating these principles to the lubrication of representative engine parts. The form of system provided to circulate oil to the various moving parts in a typical engine is then considered, together with its performance characteristics and related problems.

ELEMENTARY PRINCIPLES OF LUBRICATION

Types of Friction. The frictional conditions associated with the relative movement of two parts forced together by a load may be classified in three categories, viz., dry friction, boundary friction, and fluid-film friction. Dry friction would prevail if two chemically clean metal blocks were rubbed together. If a film of oil were spread on one of the two surfaces and then most of it wiped off, boundary lubrication conditions would prevail if the blocks were again rubbed

together. The term *boundary lubrication* is used because the thin film of oil left adhering to the surface would give only marginal lubrication. True fluid-film-lubrication conditions could be obtained by rounding the leading edge of the smaller block and then pushing it across a surface flooded with oil, as indicated in Fig. 157. If the load on the smaller block were placed aft of the center, the front would tend to ride higher than the rear. Therefore oil adhering to the mating surfaces would, through the relative motion of the two blocks, form a fluid wedge between the two parts so that the smaller block

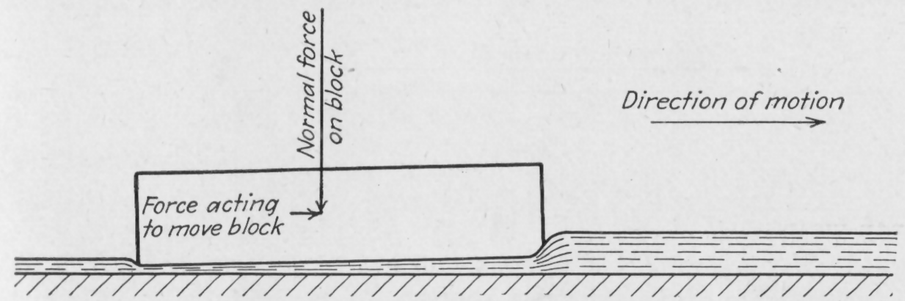


FIG. 157. Diagram of a fluid wedge built up by a block sliding across a surface covered with a thick layer of oil under conditions of perfect fluid film lubrication.

would be floated on an oil film. The effect might be compared roughly with the buoyant forces generated under an aquaplane as it is towed by a motorboat.

The coefficient of friction has been defined as the ratio of the tangential force to the normal force acting on two parts as one is moved relative to the other (see Fig. 157). While the load is usually not supported uniformly over the entire area, the load intensity is generally expressed in pounds per square inch of projected area in the regions appearing to be in contact.

The vast differences between the various types of friction can be most easily visualized by considering the coefficient of friction, the allowable bearing load, and the wear for each case. Conditions of true dry friction between metal parts rarely prevail in practice except between such parts as railroad-car wheels and brake shoes. When they are encountered, the coefficient of friction runs from 0.10 to 0.70, while the load capacity of the bearing surfaces (without rapid wear) is of the order of only a few pounds per square inch, depending upon the surface hardness. Values for boundary lubrication vary widely with the character of the surfaces, the amount and type of the lubricant, and the rubbing velocity, but a coefficient of friction of 0.02 and a load capacity of 200 psi might be considered representative for such

parts as piston rings. While the wear rate may be low, some wear invariably takes place. Under conditions of perfect fluid-film lubrication, the coefficient of friction may be as low as 0.002 and the load-carrying capacity of the bearings as high as 20,000 psi. The wear rate is zero for fluid friction, for under these conditions the solid surfaces never touch—they are always separated by the film of lubricant.

Slipper Bearings. Reciprocating parts such as pistons operate in much the same fashion as the block in Fig. 157. The relation between the load, the rubbing velocity, the oil viscosity, the minimum clearance between the parts, and the coefficient of friction can be derived

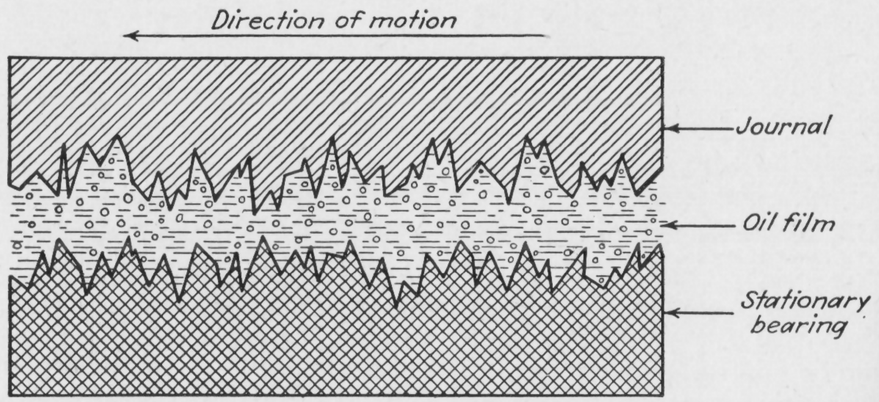


FIG. 158. Fluid wedge formed between irregular surfaces. (*The Texas Co.*)

from hydrodynamic considerations.[1] Briefly, the results of such a derivation show that, the greater the load or the lower the oil viscosity, the lower the coefficient of friction and the thinner the oil film. If the rubbing velocity is increased, the load-carrying capacity is also increased if other factors are held constant. In practice, three vital factors limit bearing loads and rubbing speeds of even the most carefully designed bearings. Probably the most important of the three factors is surface finish. If the oil film becomes thinner than the height of surface high spots caused by either waviness or roughness from machining or grinding, metal-to-metal contact will occur. A condition of this sort is indicated in Fig. 158, in which the surface irregularities are greatly magnified.[2] At low speeds metal-to-metal contact results only in rather rapid wear, but at high speeds the rate at which heat is generated by tearing of the metal in small localized areas may be so high that heat conduction out through the metal of the supporting housing will not be great enough to prevent melting of the bearing material and/or severe scoring of the bearing surfaces. The excessive heat released may also cause a part such as a piston to

expand so much that it will seize in its cylinder or a part such as a journal to seize in its bearing.

At high loads and rubbing velocities the rate of heat generation just from fluid friction with perfect fluid-film lubrication may constitute a second limiting factor on bearing capacity. The frictional heat may be evolved so rapidly that it cannot be conducted off through the parts at a rate high enough to prevent overheating. Melting of the bearing metal or seizing will result unless oil is circulated through the bearing in such quantities that it acts as a coolant and carries off enough heat to keep the surface temperatures within allowable limits.

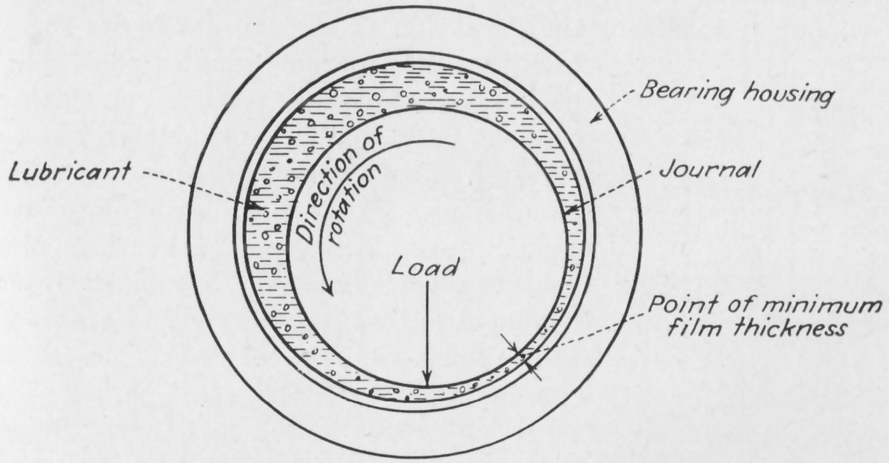


FIG. 159. Fluid wedge formed in a journal bearing. (*The Texas Co.*)

The strength of the bearing metal itself may not be adequate in heavily loaded bearings.[3–5] If this is so, fatigue cracks usually begin at the surface. These lead to other cracks under but parallel to the surface, which in turn cause spalling out of the bearing material and finally failure of the bearing when the effective area is reduced so much that the remaining surface becomes too small to support the load. A bearing that has started to fail in this way is shown in Fig. 245, p. 422.

Journal Bearings. Plain journal bearings are the most widely used of all types of bearing. The principles of slipper-type bearings can be readily extended to this type simply by allowing for the effects of the curved surfaces. Figure 159 indicates the manner in which a fluid wedge is built up in a journal bearing by rotation of the shaft. Figure 160 shows the pressure distribution in the resulting oil film. It is evident from Fig. 160 that the oil should be supplied to the unloaded side of the bearing, for it is only on that side that the pres-

sure is low enough to pump oil into the bearing clearance. In a case such as that of a crankshaft in which the loads rotate with the shaft so that they are applied—in this case—mostly on the inner sides of the crankpins, the oil may be supplied through holes in the shaft to the unloaded side of the journal.

The diametral clearance is the most important characteristic of a journal bearing if a good surface finish is presupposed. The greater the diametral clearance, the smaller the area effective in carrying the load and hence the lower the bearing capacity for a given oil viscosity and surface finish. On the other hand, if the diametral clearance is too small, the oil-flow rate through the bearing may not be adequate for cooling purposes or the expansion of the journal under running conditions may become greater than that of the bearing and cause seizing. A clearance of about 0.001 in./in. of diameter has been found to be satisfactory for most automotive applications. Production techniques have made it possible to produce bearings of the requisite surface finish and to sufficiently close tolerances so that just the right clearance and surface finish can be provided at the factory. Thus a gradual "running-in" period at low speeds and loads is not necessary. Because of this, since the early 1930's the buyer of a new car has not been required to drive slowly for the first 1,000 miles or so to wear excess material from tight bearings and thus obtain adequate clearances for high-speed operation.

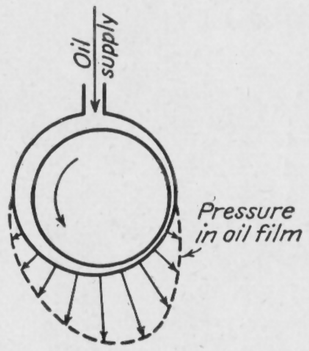


FIG. 160. Pressure distribution in a journal bearing.

Effects of Speed and Load on Journal-bearing Performance. The widespread use of journal bearings with provision for full fluid-film lubrication makes the performance characteristics of this element of a lubrication system of particular interest. The effects of load on the coefficient of friction, the frictional power losses, the oil flow rate, the bearing temperature, and the consequent oil viscosity are shown in Fig. 161 for an application requiring a highly developed, high-capacity type of journal bearing.[6] The effects of shaft speed on the same factors are given in Fig. 162 for a constant bearing load. Note that the coefficient of friction increases only slightly with speed because of the drop in viscosity occasioned by the higher bearing temperatures. Note, too, the gradual increase in oil flow rate together with the slightly greater than linear power loss with an increase in speed, and much the same characteristics for an increase in load.

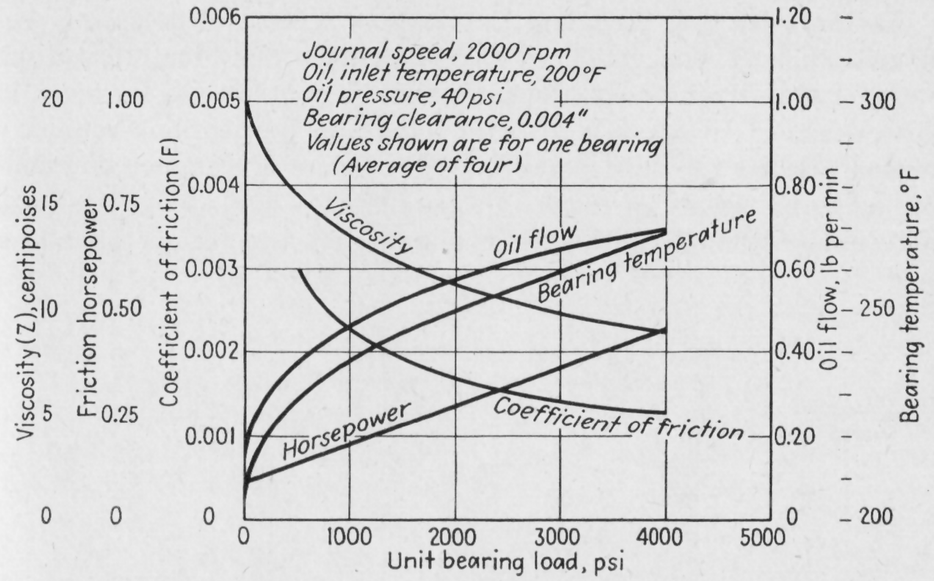


FIG. 161. Effect of load on factors affecting the performance of a 3.25-in.-diameter journal bearing. (*Palsulich and Blair, Trans. SAE, Vol.* 54, 1946.)

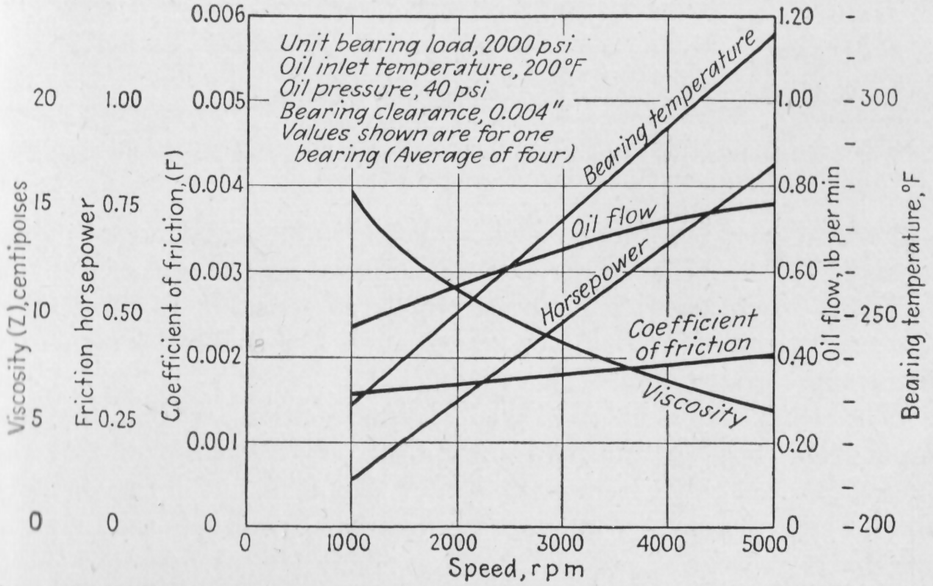


FIG. 162. Effect of rpm on factors affecting the performance of a 3.35-in.-diameter journal bearing (*Palsulich and Blair, Trans. SAE, Vol.* 54, 1946.)

Factors Affecting Boundary Lubrication. The two preceding sections have been concerned only with perfect fluid-film lubrication. Boundary-lubrication conditions must also be considered, for, even in parts designed to operate with full fluid-film lubrication, boundary friction prevails for short periods either because of a momentary failure in the oil supply or during a start before a fluid wedge has been built up. Although it might be hoped that the former condition

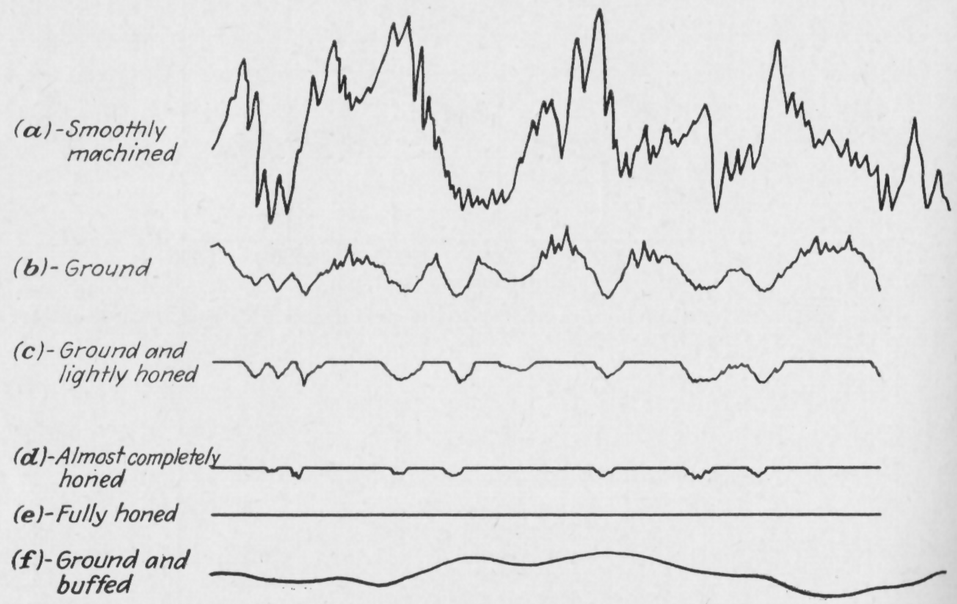


FIG. 163. Representative types of surface irregularities—vertical dimension magnified 1,000 times, the horizontal dimension magnified 10 times.

could be avoided, the latter condition is inevitable and allowance must be made for it. The former really cannot be ignored either, for few operators would care to have an engine so sensitive to lubrication that bearing failures would be threatened by the slightest irregularity in the rate of oil supply.

The requirements for good boundary lubrication are somewhat different from those for full fluid-film lubrication. Again, probably the most important single item is that of surface finish, but in this instance in a different way. A wide variety of surface structures are found in practice.[7] A bearing surface that has many tiny depressions interspersed between high spots with relatively flat tops, as shown for the surface which has been ground and honed, of Fig. 163*c*, has been found to give the best performance. Not only do the irregular depressions aid the oil film to adhere to the bearing, but they provide a multitude of scattered reservoirs from which oil may be continually drawn

to lubricate the high spots. In engines, the effects of both thermal distortion and deflection under load are sufficiently great to cause high spots to form and hence give boundary-lubrication conditions where they would not otherwise occur. Whenever this condition is likely to obtain—and it is likely with practically every engine part—the bearings must be designed so that portions of the surfaces may operate under boundary-lubrication conditions for appreciable periods. Because of this it has been found that exceptionally high surface smoothness is actually undesirable for most engine parts, for two such surfaces tend to squeeze all the oil out from between them and then weld together. A surface roughness of about 60μ in. has been found optimum if the finishing process yields the sort of surface shown in Fig. 163*c*. Other surfaces of the same highest-peak-to-lowest-valley roughness but of the sort shown in Figs. 163*b* or 163*f* are much less desirable.

The type of molecules in any given lubricant affects its lubricating value under conditions of boundary friction by their tendency to adhere to metal surfaces.[8] Some liquids such as the silicones are not inclined to "wet" some surfaces and hence may be easily rubbed or squeezed from the surfaces of moving parts. Others such as animal and vegetable oils contain (or are wholly made up of) molecules that are "polarized" in that each carries an electrostatic charge at one end in the form of an OH group, for example. The ends of these polarized molecules tend to attach themselves to metal surfaces, thus forming closely packed and very adherent layers of lubricant molecules. It has been found that only small percentages of such polarized compounds need be added to a mineral oil to give close to the maximum benefits obtainable; for the polarized molecules are attracted to metal surfaces, and only a small percentage of the number of molecules in the lubricant is needed to form the desired layers.

Both the surface hardness and the melting point of a material affect its suitability for use under boundary-lubrication conditions. A plating of a soft material such as tin or lead may be applied to a surface such as the face of a piston ring. In the event of a local breakdown in lubrication the tin will melt and serve as a lubricant. The wear rate, of course, is inevitably high. A very hard surface coating may also be applied to correct a boundary-lubrication problem. To take piston rings as an example again, hard chromium plate has been applied to the ring face to correct trouble with ring scuffing. Since the hard material greatly reduces the wear rate, it is often preferable as a means of correcting boundary-lubrication trouble.

The bearing loads under conditions of boundary lubrication are usually not high enough to make the fatigue strength of the bearing metal important. It is important, however, that the bearing alloy should not tend to weld to the shaft. That is, if a steel shaft is used, the bearing should not be of steel but rather should be of babbitt metal, bronze, aluminum, cast iron, or other material that will not tend to weld to the steel if a partial breakdown of lubrication occurs. It is true that, for some applications in which the rubbing velocities have been low, nitrided steel has been successfully used in contact with a carburized-steel part; but even in such a case the materials are really dissimilar, for the surface coatings of the two parts are different.

The tendency of two surfaces to weld together can be reduced by the use of special lubricants. As mentioned in Chap. VIII, sulphur, phosphorus, or chlorine compounds may be added to oils so that they will form surface coatings that will prevent welding. These coatings not only prevent welding, but the more effective have a pronounced lubricating value within themselves in that they are soft and act like graphite, shearing easily under load. Since these sulphur, phosphorus, and/or chlorine compounds are usually by nature mildly corrosive, care must be exercised in their selection and use to make certain that the coating formed will be "protective," *i.e.*, that the coating formed will prevent the active material from reaching the metal beneath the coating. Thus the additive will attack the metal only where friction has stripped the coating from a small portion of the rubbing surface and left the bare metal exposed. Lead soaps are also sometimes used in these lubricants because of their high load-carrying capacity.

Lubricants of this type are called *extreme pressure* (EP) lubricants. They have proved to be spectacularly successful in some applications, more than doubling the load-carrying capacity of hypoid gears for rear axles, for example.

Lubrication of Gear Teeth. A considerable amount of relative sliding motion occurs at the line of tooth contact in gears. The problems are usually not great in low- or medium-speed applications; for unless the loads are very high, the gears may be operated so that they dip into a bath of lubricant. This is the arrangement commonly employed in automobile transmissions and differentials. At high speeds, however, it is important that the supply of lubricant be limited, especially if the tooth height is small compared with its length. The reason for this is that the oil trapped between engaging teeth must be forced out as the teeth move from the point of initial contact to the point of full engagement. This "forcing out" may cause high pressures to be built up in the trapped oil and thus push the gears apart and increase the backlash. Such a condition is known as *hydraulic loading* of gear teeth. It is accompanied by a power loss, high bearing loadings, and excessive heat generation, which may damage the parts. The formation of many small air bubbles that remain entrained with the lubricant and tend to fill the entire gearbox with foam usually accompanies and aggravates this condition.

One of the most successful high-speed-gear-lubrication devices has been an oil jet directed into the gear teeth where they disengage. This serves both to keep them cool and to maintain a thin film of lubricant on the teeth. Excess lubricant is thrown off before the teeth have rotated to the point where they again engage.

Ball and Roller Bearings. The lubrication requirements of ball and roller bearings are quite different from those for slipper or journal bearings. Some rubbing occurs between the balls or rollers and the retainer, or separator, that keeps them properly spaced, but the loads are so small that little heat is generated and boundary lubrication from a thin oil film is entirely adequate. A slight amount of rubbing under heavy loads does occur between the balls or rollers and the races as these parts distort in the vicinity of the contact area. The effect is analogous to the deflection in a rubber tire as it rolls along a road and the scrubbing action between the tread and the road that results. While the deflections are primarily in the race of a bearing rather than in the rollers, the effect is the same in that relative surface movement under load occurs in both cases. The amount of this slippage in bearings is so small, however, that a slight film of oil is adequate. In fact, a flood of oil not only contributes no improvement to high-speed-ball- or roller-bearing lubrication but is actually a detriment, for the oil is churned so violently by the rapidly moving parts that a considerable amount of heat is generated. Not only does the heat represent a power loss, but it also raises the operating temperature of the bearing, tends to anneal the hardened steel surfaces, and thus shortens bearing life. The best form of ball- or roller-bearing lubrication is provided by a fine mist of oil or an occasional but regular droplet, which the rapidly moving bearing quickly converts into a mist.

ENGINE-LUBRICATING SYSTEMS

Types of Lubricating System. Many means have been devised for supplying lubricant to the rubbing surfaces in engines. Probably the simplest is that employed in outboard-motorboat and model-airplane two-cycle engines—oil is mixed with the gasoline in the ratio of

about 1:10 and supplied with the fuel. This arrangement is well suited to these engines because the air is induced from the carburetor into the crankcase and compressed there before delivery to the cylinders. All the principal moving parts are exposed to the wet mixture and continually have a fresh supply of lubricant deposited on them. Such an arrangement is wasteful of lubricant and is not adequate for any but this simple type of engine. The famous model T Ford together with most of the other engines of its day employed a *splash* lubrication system. The oil in the crankcase was maintained at a level such that each connecting rod dipped into the oil bath at the bottom of its stroke. A scoop, or dipper, attached to the bearing cap scooped up oil, the force of the impact carrying the oil through a hole to a groove in the bearing surface. The pistons and cylinder walls were lubricated by splash, while drain passages were designed to carry oil to the main bearings, the camshaft, and the valve gear. Although this type of system was adequate at low speeds, as engine speeds and outputs were increased it was found that oil churning in the crankcase became severe and caused an excessive power loss. Even more important, the oil flow rate to the main- and connecting-rod bearings was not sufficient to carry off the frictional heat rapidly enough to maintain bearing temperatures at safe values. The next logical step was to provide a small oil pump and a set of passages to give a positive supply of oil to the main bearings, the camshaft, and the valve gear. The Chevrolet engines in production at the time of writing have a lubrication system of this type except that the connecting rods dip, not into the oil reservoir in the crankcase, but instead into troughs mounted in the crankcase above the normal oil level. An oil jet located in each trough is directed in such a way that it is intercepted by the dipper on the connecting rod. This at once provides a more positive supply of oil to the rod bearings and greatly reduces oil-churning power losses. In most engines in recent years a further step has been taken and use has been made of drilled passages in the crankshaft to carry oil from grooves in the main bearings to the rod bearings. An arrangement of this sort may be seen in Fig. 164. Since this type of lubricating system is so widely used, it will be described in some detail.

A Full-pressure Lubricating System. The relative simplicity, exceptional reliability, and adaptability of the full-pressure oil lubricating system have been such that it is now used in virtually all engines that develop more than about 30 hp. Its basic elements are shown in Fig. 165. A gear-type pump draws oil through a screen

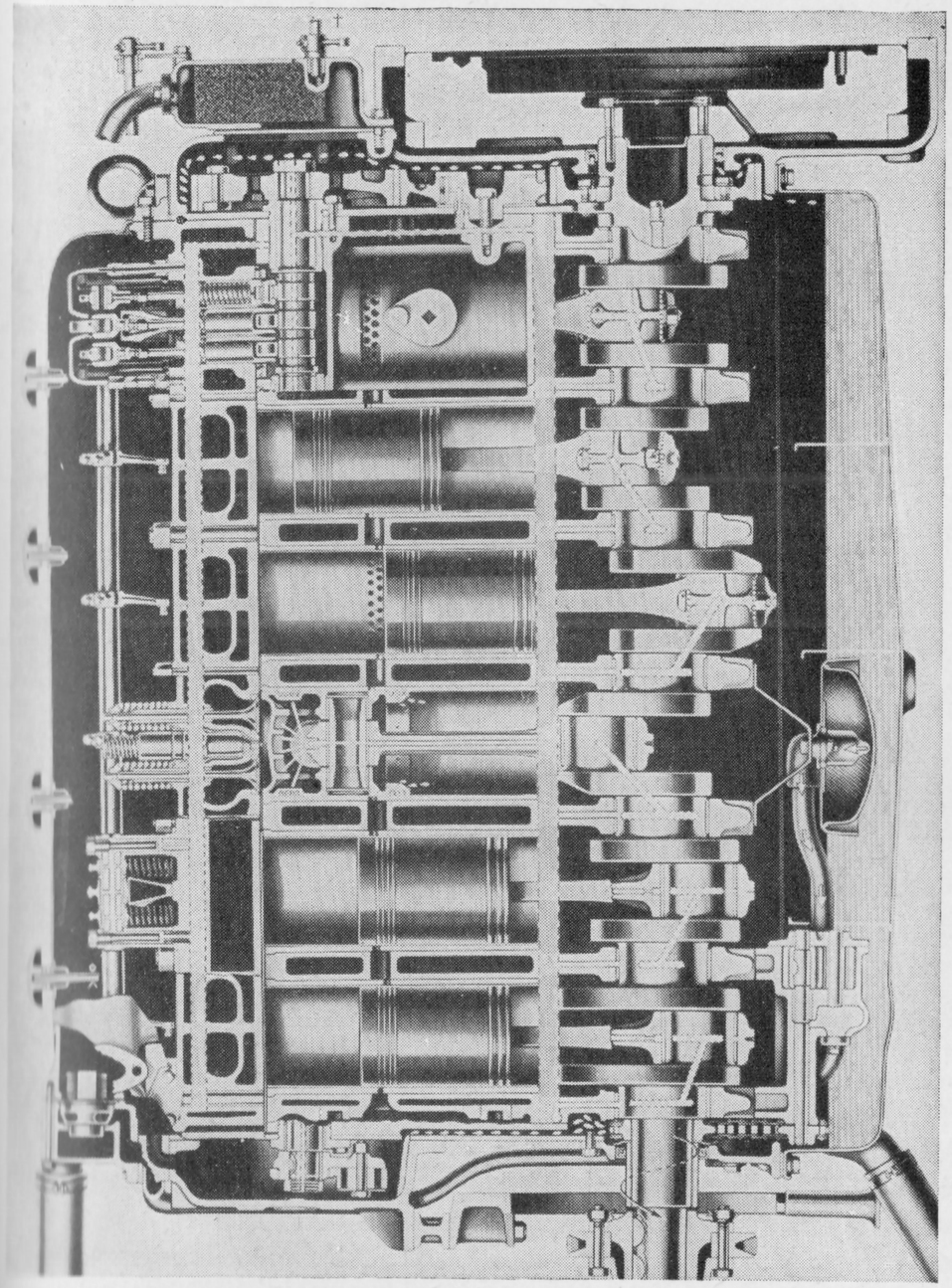


FIG. 164. Longitudinal cross section through a six-cylinder truck diesel engine showing the flow of oil from the pan through a floating strainer to the pump and thence to the passages supplying the main bearings, connecting rods, pistons, valve gear, etc. *(Detroit Diesel Engine Division, General Motors Corp.)*

from the oil reservoir. The pressure of the oil delivered to the engine is maintained at an essentially constant value by an oil-pressure relief valve. The relief valve is necessary to prevent excessive pressures and possibly failure of the pump under cold-starting conditions, when the oil is very viscous. The oil flows from the pump through passages to the various surfaces requiring lubrication. These passages are

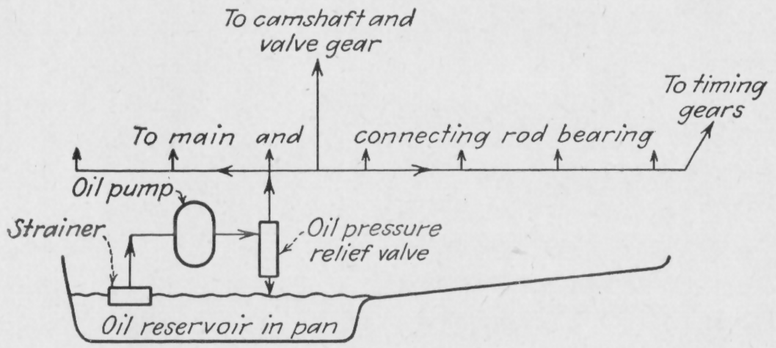


FIG. 165. Simple full-pressure engine lubrication system.

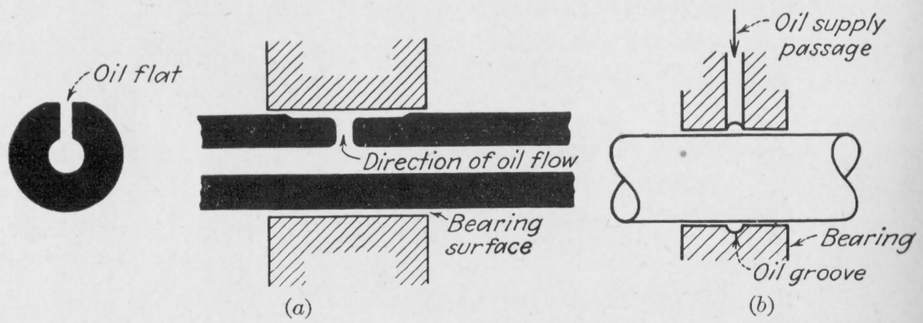


FIG 166. Devices often employed to supply oil to journal bearings. (*a*) Oil flat on shaft, (*b*) Circumferential oil groove in a bearing. Note that the oil flat is terminated *inside* the bearing to prevent oil leakage out through the ends.

usually cored or drilled in the engine housings, although steel or copper tubes are also often employed where internal passages cannot be provided conveniently. The passage size is generally such that the major restriction to oil flow is where the passage terminates at a bearing, the clearance space between the journal and the bearing largely determining the oil flow rate to that part. Oil flats or grooves are often provided in the journal or the bearing, both to distribute the oil uniformly over the rubbing surfaces and to prevent the annular orifice formed at the oil-passage terminus from becoming a severe restriction to oil flow. These devices are shown in Fig. 166. Oil flats are cut in the shaft in instances where the oil is supplied from the inside of the shaft, while oil grooves are cut in the bearing in instances where

the oil is supplied through that part. It should be emphasized that oil grooves in bearings should be very carefully designed as they may greatly reduce the load capacity of the bearing. This effect can be visualized easily by considering the effect of an oil groove in the high-pressure region of the bearing in Fig. 160. The groove would act to vent the oil away from the high-pressure region and allow the two metal surfaces to come together. Circumferential grooves of the type shown in Fig. 166 have been found to be, in this respect, the least offensive type of groove. They are often employed to provide a continuous oil passage from a stationary part through a hole to the inside of a rotating part. Crankshaft main bearings, for example, are grooved (see Fig. 164) so that oil under pressure is maintained in an annulus completely surrounding the journal. This makes possible a continuous supply of oil through the drilled passage in the crankcheek to the connecting-rod bearing. Without the groove in the main bearing, the end of the oil passage through the crankcheek to the crankpin would register with the main-bearing oil-supply hole for only a few degrees each revolution, and hence the oil supply to the rod bearing would be severely restricted. A grooved bearing is also often employed at the middle or the end of the camshaft so that oil can be supplied from the crankcase to the center of the long, hollow camshaft, from which it can be bled out through holes in the journals to lubricate the various bearings along the shaft. Valves, valve guides, and valve springs may then be lubricated by splash from the camshaft bearings and tappet guides.

The pistons, piston rings, and cylinder walls receive their oil supply from the lubricant thrown off by the connecting-rod bearings. This splash oil is distributed over the piston and piston-ring surfaces by the slight oscillating motion of the piston as it reciprocates in the cylinder, acting as a "slipper" bearing of the type indicated in Fig. 157.[9]

The oil pan must be so designed that the oil level is always well below the crankshaft counterweights and the connecting rods, for the high air velocities induced by these parts tend to churn and whip the oil. Once started, such a condition tends to grow worse very rapidly as air bubbles become entrained in the oil. These raise the effective oil level in the pan and fill the crankcase with foam. Oil loss through the breather is an indication of this condition.

The oil pump is usually designed to give a higher flow rate than the engine will require at normal speeds so that there will generally be some by-pass flow through the valve and hence effective pressure

regulation. For this reason, if the engine oil pressure falls below its normal value (25 to 80 psi for most engines) at moderate or high speeds, serious trouble has probably arisen. The most probable cause is a low oil supply, *i.e.*, that the oil level in the reservoir has dropped so low that the pump is sucking in air. Another possibility is that a bearing has failed and is allowing oil to flow freely out the end of some passage. Oil aeration, or foaming, as mentioned in the previous paragraph will also cause oil-pressure fluctuation. Because of its sensitivity to such disturbances in the lubricating system, the engine oil-pressure gauge is probably the most important instrument to be watched while an engine is being started or operated.

Effects of RPM on Oil Pressure and Flow Rate. The effects of engine speed on the oil pressure and the flow rate to the bearings are shown in Fig. 167 for a representative light-truck engine. Ideally, the output of the oil pump should be directly proportional to speed. The actual output will run somewhat below this because of back leakage through the internal pump clearances. As indicated in Fig. 167, the percentage loss from this effect is particularly high at low speeds. It is also evident in Fig. 167 that the engine oil pressure depends on the rate at which the oil escapes through the bearings at low speeds. This dependency of oil pressure on flow rate through the bearings obtains until the speed is increased to the point where the pump delivers enough oil so that the pressure drop through the oil passages and bearings exceeds the oil-pressure relief-valve setting. Oil by-pass through the relief valve at higher speeds serves to maintain a constant oil pressure, while the oil-flow rate through the bearings increases gradually, as indicated by both Fig. 162 and Fig. 167. If desired, the oil-flow rate can be calculated for a proposed design by using the simple basic relations of elementary fluid mechanics.[10]

If an engine is cold, the output of the oil pump will be slightly greater than for the warmed-up condition represented by Fig. 167—particularly at low speeds. This is a consequence of reduced back leakage. At the same time, engine temperature has a marked effect on the oil flow rate through the bearing clearances, the higher oil viscosity greatly reducing the oil flow rate. Thus, while an engine is being warmed up, the oil pressure under idling conditions will run higher than it will at the same speed after the engine has reached normal operating temperatures.

Oil Filters and Strainers. In the larger and more highly developed engines, it has been found very much worth while to include additional special provisions to keep the oil clean. Filters may be

added to remove fine particles of dirt (siliceous material is particularly objectionable) and thus reduce engine wear. The most common arrangement is the *by-pass filter*, which handles only a portion of the oil that passes through the oil pump during any given cycle. That

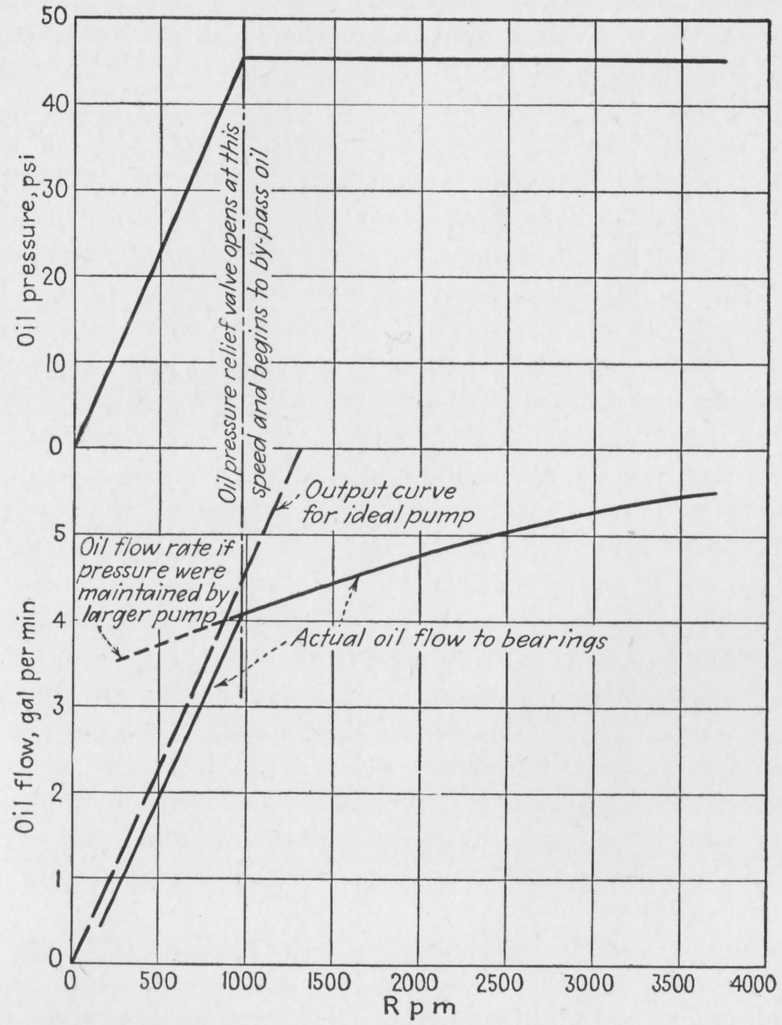


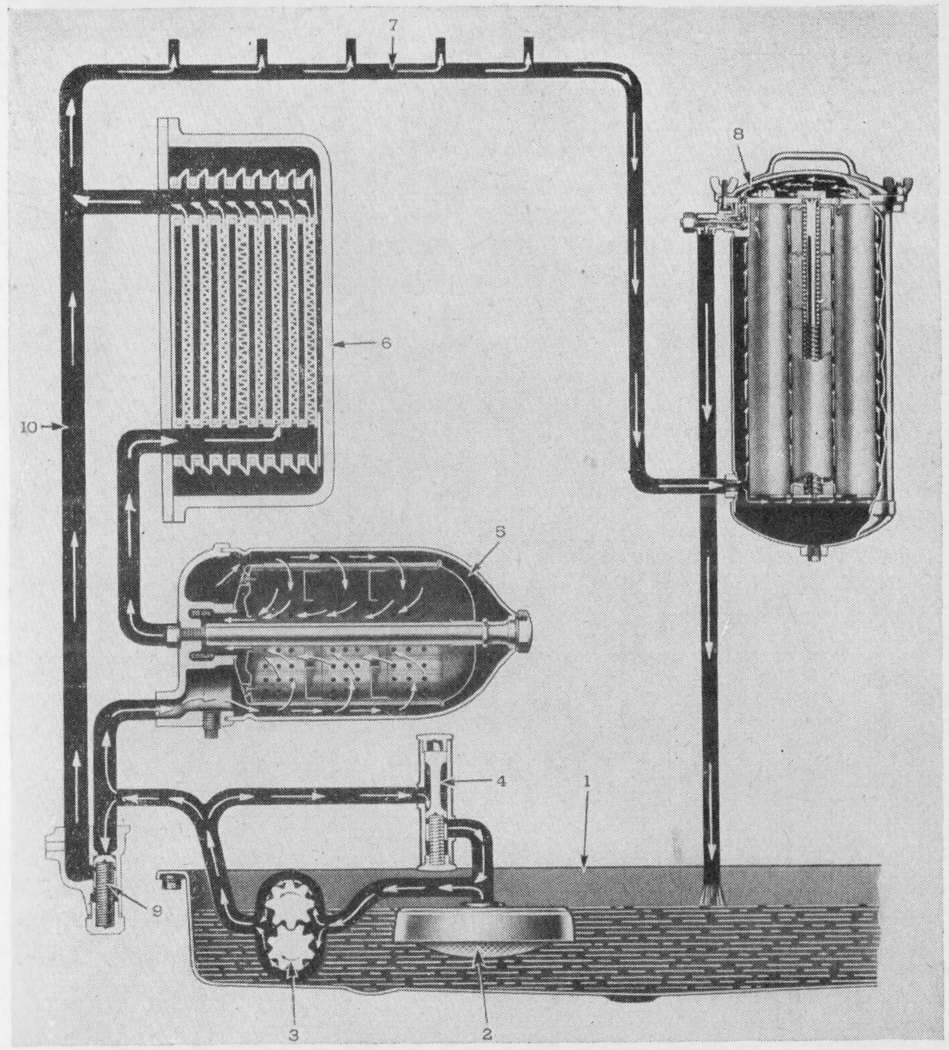
Fig. 167. Effect of engine rpm on oil-flow rate and oil pressure for a light truck engine when fully warmed up.

is, perhaps 80 per cent of the oil delivered by the pump will flow to the engine bearings. The balance may be by-passed through the filter to the crankcase. A system of this type is shown in Fig. 168. A full-flow oil filter is sometimes employed, but it must be large and hence bulky if it is to be effective and yet not clog rapidly. Since such a filter must be placed between the pump outlet and the engine, the engine oil pressure will fall off if the pressure drop across the filter

becomes excessive. Thus a by-pass valve must be provided with full-flow filters to safeguard the engine. Because of the difficulty involved in filtering very fine particles from the oil and since the larger particles cause the greater part of the wear, fine strainers have been used in many engines and have proved to be very successful. Strainers of this type may remove practically all dirt particles down to sizes as small as 0.003 in. in diameter. As is the case with full-flow filters, a by-pass valve is usually provided around the strainer so that the oil pressure will always be maintained even if the strainer clogs, for a plentiful supply of dirty oil is far better than an inadequate supply of clean oil. Very often an oil system may employ both a full-flow strainer and a by-pass filter. Such a system is shown in Fig. 168.

Oil Coolers. In most passenger cars and in the smaller trucks and busses, air circulation about the crankcase caused by the motion of the vehicle is depended upon to keep the oil cool. While this is fairly satisfactory under most conditions, in hot weather crankcase oil temperatures in some types of engine may reach values as high as 350°F. Since any increase in oil temperature causes a rapid increase in the oil-deterioration and bearing-corrosion rates as well as excessive engine deposits and high piston-ring wear rates, it has been found desirable—and sometimes essential—to provide an oil cooler. The usual procedure is to place a heat exchanger between the oil leaving the oil pump and the engine coolant leaving the radiator. Such an arrangement has the advantage that the oil is warmed up quickly and kept warm in winter under light-load conditions and yet kept from getting too hot in the summer. The oil system of Fig. 168 includes an oil cooler of this type. Similar arrangements are often made for the smaller and medium-sized stationary and marine engines. Engines of more than 500 hp almost always have a separate supply of coolant for the oil cooler.

Breathers. Closely associated with the problem of engine lubrication is that of crankcase breathing. The crankcase is carefully closed off and sealed both to prevent oil from splashing or leaking out and to prevent the entry of dirt. However, a passage must be provided to permit the escape of the gases that leak past the piston rings. These gases contain water vapor from the products of combustion, together with traces of sulphur and/or lead compounds from the gasoline. These compounds when dissolved in condensed water vapor are very corrosive. Thus some means of crankcase ventilation is essential, both to prevent the blow-by gases from building up pressure in the crankcase and hence causing oil leaks through shaft seals and parting surfaces and to remove the potential corrosion hazard inherent in these gases. The usual procedure is to provide a breather tube that taps into the crankcase at some point relatively free of splashing oil. Baffled passages in the breather tube or cap permit gases to escape and yet intercept small oil droplets and allow them to drain back into the crankcase. A gentle but definite circulation of air through the crankcase may be provided to give positive scavenging of these gases but is not essential if the crankcase oil temperature is



1. Oil pan
2. Screen—oil-pump intake
3. Pressure oil pump
4. Oil-pressure relief valve
5. Oil strainer
6. Oil cooler
7. Oil manifold in cylinder block
8. Oil cooler
9. Strainer and cooler by-pass valve
10. Strainer and cooler by-pass passage

FIG. 168. Schematic diagram showing the path of oil through a lubrication system having a single pressure pump, strainer, cooler, cleanable element filter, and single sump oil pan. (*Detroit Diesel Engine Division, General Motors Corp.*)

kept above 180°F so that condensed fuel and water vapor will be evaporated off.

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Problems

1. Distinguish between fluid-film, boundary, and dry lubrication.

2. List three conditions under which a bearing that normally operates with fluid-film lubrication might have to operate, at least in part, with boundary lubrication.

3. Given three classes of surface roughness, group them in the order of their desirability, first for boundary lubrication and second for fluid-film lubrication. The three classes of surface roughness are as follows: (*a*) optically flat, (*b*) 8 to 10 μin., and (*c*) 0.0001 to 0.001 in.

4. What effect does surface hardness have on allowable loads under boundary-lubrication conditions?

5. List four distinctly different ways in which bearing failures may occur in properly aligned bearings supplied with a continuous flow of lubricant.

6. Sketch an engine-lubrication system labeling the following: oil pump; pressure relief valve; strainer; filter; main- and connecting-rod bearings; valve gear.

7. State a common specific detail function of each of the following in an engine: oil flat; oil groove in a bearing; oil jet; oil gallery.

8. Explain briefly why the oil pressure of an engine might remain constant during the initial portion of the warm-up but fall off when the engine reaches its normal operating temperature.

9. What effect would increasing the tension on the oil-pressure relief-valve spring have on the engine of Prob. 8?

10. Sketch curves of both oil flow rate through an engine and oil delivery rate from a gear-type oil pump as functions of engine rpm. Assume a constant oil pressure and temperature in both cases.

CHAPTER XI

COOLING

Internal-combustion engines must be cooled, or the lubricating-oil film between the moving parts will break down, and damage to the engine will result. While other factors such as the loss in strength of metals at elevated temperatures, thermal distortion, and combustion problems such as preignition or detonation may sometimes become critical before a lubrication failure occurs, lubrication imposes temperature limits that may not be exceeded in any engine for any condition of operation. On this basis, the foremost function of the cooling system is to maintain the moving parts at the optimum temperature for lubrication. If this condition is satisfied, the other requirements are usually not difficult to meet. To do this, the engine cooling system must carry off and dissipate an amount of energy roughly equivalent to the bhp output of the engine.

The principal elements of the engine-cooling process are as follows: (*a*) heat transfer from the gases in the combustion chamber and exhaust port to the cylinder head and cylinder walls, together with heat conduction through the metal engine parts to the coolant; (*b*) heat transfer from the engine parts to the coolant, coupled with coolant flow and circulation; (*c*) the characteristics of radiators for liquid-cooled engines. Since a basic understanding of heat transfer is essential for a proper approach to the various phases of engine cooling problems, a brief discussion of the necessary fundamentals is given as an introduction. An additional section on the specialized practical problems concerning both the design and the maintenance of liquid-coolant systems is given in the last part of the chapter.

BASIC HEAT-TRANSFER RELATIONS

Heat Flow. It is a principle of elementary physics that the rate at which heat will flow through a metal part is directly proportional to the flow-path area, the temperature difference between the hot and cold extremities, and the heat conductivity of the metal, while it is inversely proportional to the length of the heat flow path. This may be expressed algebraically as follows:

$$Q = \frac{kA\ \Delta t}{l} \tag{13}$$

where Q = heat-transfer rate, Btu/hr
k = heat conductivity, Btu/(hr)(ft²)(°F)(ft)
A = heat flow path cross-sectional area, ft²
Δt = temperature difference, °F
l = length of heat flow path, ft

Heat transfer through the film between a solid and a fluid follows the same relation except that the conductivity and the length of the heat flow path are replaced by the surface heat-transfer coefficient h.

$$Q = hA\,\Delta t \tag{14}$$

This formula is very important, for it defines one of the most fundamental and widely used relations encountered in mechanical-engineering work. The effective heat-transfer area and the temperature difference are simple independent variables, but the heat-transfer coefficient is a complex quantity that depends on many factors.

Surface Heat-transfer Coefficient. The most important barriers to the flow of heat are the surface films between solids and fluids. The resistance to heat flow of such a film is determined by both its effective thickness and the heat conductivity of the fluid. The effective thickness depends upon the velocity of the fluid flowing over the surface and the type and amount of turbulence. This, in turn, is dependent on the viscosity of the fluid and the diameter, or thickness, of the fluid stream. The density and the specific heat of the fluid are also important factors as they determine the amount of heat which will be picked up by those particles which circulate into and out of the surface film. A more detailed discussion of the problem is beyond the scope of this book, but there is ample theoretical and experimental proof to show that, under turbulent-flow conditions, the following relation holds for any fluid within the limits of experimental error:[1]

$$h = \frac{26.4(k)^{0.6}(c_p)^{0.4}(G)^{0.8}}{(\mu)^{0.4}(D)^{0.2}} \tag{15}$$

where h = heat-transfer coefficient, Btu/(hr)(ft²)(°F)
c_p = specific heat of fluid at constant pressure, Btu/(lb)(°F)
k = thermal conductivity of fluid, Btu/(hr)(ft²)(°F)(ft)
μ = viscosity of fluid at bulk temperature, lb/(hr)(ft)
G = fluid-flow rate, lb/(hr)(in.² of flow-passage area)
D = diameter of flow passage, in.

For a given engine operating with a particular coolant, the only variable in Eq. (15) is the fluid flow rate G because c_p, k, and μ remain essentially constant over the narrow temperature range of interest.

Therefore Eq. (15) may be rewritten as

$$h = K(G)^{0.8}$$

where K is a constant depending upon the coolant and the geometry of the coolant flow passages. Figure 169 was computed from Eq. (15)

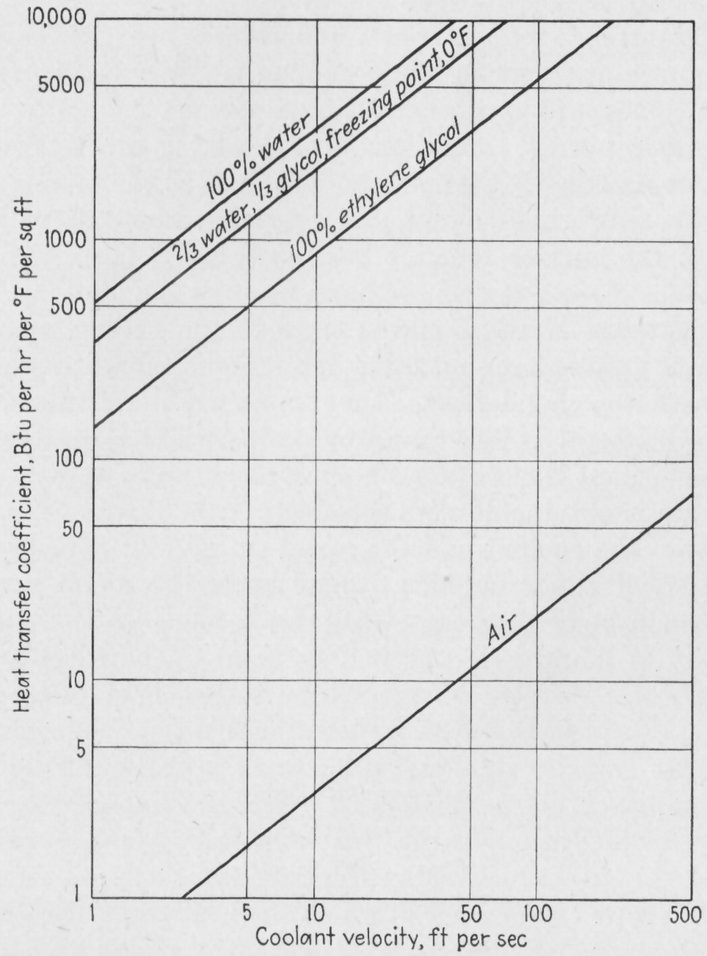

Fig. 169. Effect of coolant velocity on the surface heat-transfer coefficient for flow through smooth tubes 1 in. in diameter. Computed from Eq. (15).

to give some notion of the relative values of this heat-transfer coefficient for three of the fluids used for engine cooling. Water has the highest heat-transfer coefficient at any particular flow rate, glycol gives a value about one-fourth as great principally because of its higher viscosity, and air has a heat-transfer coefficient about one one-thousandth of that for water, largely because of its low density. On this basis alone, it is evident that it is more difficult to obtain satisfactory cooling with glycol than with water and much more difficult

to obtain good cooling with air than with either glycol or water. Higher coolant flow rates may be employed with glycol to offset its lower heat-transfer coefficient, but air cooling requires not only much higher coolant velocities but much greater cooling-surface areas in the form of the cooling fins that make the cylinders of air-cooled engines so easy to identify (see Fig. 27).

Heat Transfer to a Coolant at Temperatures near Its Boiling Point. A very important factor in liquid cooling is the heat-transfer mechanism that comes into play when a liquid coolant is close to its boiling point. Under normal conditions, the liquid contacting the metal surface is at practically the same temperature as the metal; and if the heat transfer rate is high, a high temperature gradient prevails through the rest of the surface film, or boundary layer, between this inner layer of the surface film and the main body of the liquid. When the bulk temperature of the liquid is close to the boiling point, a high temperature gradient cannot exist because the liquid immediately in contact with the metal boils. In vaporizing, it abstracts far more heat from the metal surface than would otherwise be possible. Therefore, if the coolant is close to its boiling point, an extremely high heat transfer rate is obtainable with relatively little difference in temperature between the coolant and the metal surface. Under these conditions the small vapor bubbles formed leave the metal surface, mix with the main body of the coolant, and recondense there unless the entire body of liquid is at the boiling point. The excellent wetting action of water coupled with its high latent heat of vaporization make it an exceptionally good coolant for this type of cooling.

The heat transfer rate prevailing with a particular liquid under boiling conditions has been found to depend almost entirely on the temperature difference between the hot metal surface and the boiling point of the liquid[1] (which is, in turn, dependent only on the pressure). Figure 170 shows the manner in which the heat transfer rate increases with the temperature difference between the metal and the boiling point of the coolant. It also shows the very important fact that a peak heat transfer rate is reached beyond which bubbles are evolved so rapidly from the hot metal surface that they form a vapor blanket which tends to prevent the liquid from touching the metal. An abrupt drop in the heat transfer rate accompanies the condition. Figure 170 shows also that scale deposits on the metal surface simply increase the temperature difference between the metal and the coolant at a given condition—the temperature drop through the scale being responsible. Since under boiling conditions the vital part of the heat-

transfer mechanism is in the liquid layer immediately adjacent to the metal surface, the heat transfer rate is not affected appreciably by coolant velocity or turbulence.

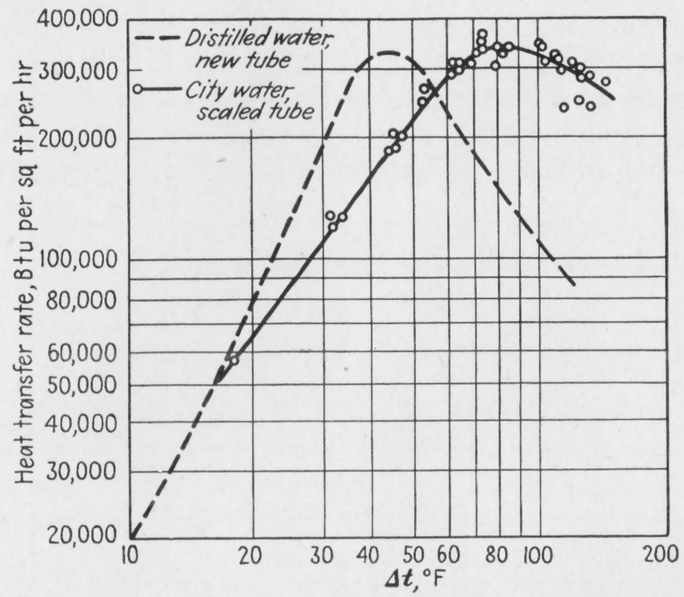

FIG. 170. Effect of heat-transfer rate on the temperature difference between a heated metal surface and boiling water. (*McAdams, "Heat Transmission," New York, N.Y., 1942.*)

HEAT LOSS TO THE COMBUSTION-CHAMBER WALLS

From Eq. (14) it may be deduced that the heat loss to the combustion-chamber walls depends upon the temperature of the hot gas in the combustion chamber, the combustion-chamber-wall temperature, the combustion-chamber-wall area, and the surface heat-transfer coefficient. To simplify the discussion, the effects of each of these factors will be considered separately. The effects of operating variables important in practice then follow directly.

Gas Temperature. The temperature of the gas in the combustion chamber increases from a value a few hundred degrees greater than atmospheric during the intake stroke to around 4500°F at the end of combustion and then drops to about 1700°F in the exhaust port in spark-ignition engines.[2] The mean effective gas temperature has been found to be about 1200°F in spark-ignition engines.[3,4] As may be deduced from the derivation on page 35, the temperature at any point in the cycle of a spark-ignition engine is almost unaffected by either throttling or supercharging. The fuel-air ratio, however, does have a pronounced effect on the gas temperature if the mixture is

considerably richer or leaner than the chemically correct value. This effect may be seen in Fig. 46. It is particularly important in diesel engines, for their power output is controlled by reducing the fuel flow from the full-load value without changing the air flow. Thus the mean gas temperature increases linearly with load in the diesel.

Combustion-chamber-wall Area. The top of the piston and the interior surface of the cylinder head are exposed to the hot gases of combustion throughout the power and exhaust strokes. The amount of exposed area in the cylinder head for a given diameter of cylinder

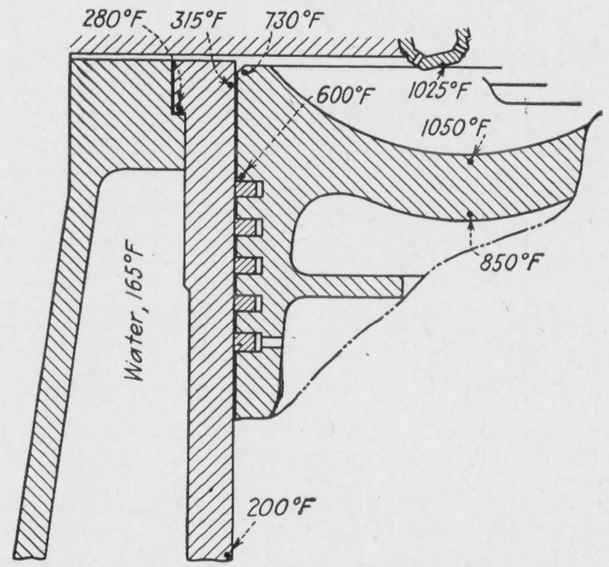

FIG. 171. Temperature of various parts in a diesel-engine cylinder. (*Rosen, Trans. SAE, Vol.* 32, 1937.)

varies with the type of combustion chamber, being higher in an L-head than an I-head engine, for example. The exhaust-port walls are exposed to hot gases moving at a high velocity during the exhaust blowdown and the exhaust stroke. The high gas velocity causes a substantial amount of heat to pass into the port walls, from which it must be carried off by the coolant. The cylinder wall area exposed to the hot gases varies with piston position. The bulk of the heat that passes into the top of the piston also flows through the piston rings into the cylinder walls. While the relative proportions of these various areas vary from one engine to another, in any given engine they are constant and a single equivalent effective area can be employed.[2]

Combustion-chamber-wall Temperature. The temperature of the metal parts that enclose the combustion chamber varies with the length and resistance of the heat flow path, the rate of heat flow, and

the temperature of the coolant. Figure 171 shows the temperature of representative points in the cylinder of a diesel engine. The highest temperature shown is in the center of the piston. The reason for this is that the heat flow path to the cooling water is longer for that point than for any other point given. The temperature at the tip of the injection nozzle is nearly as high in this instance because there are a number of surface films through which the heat from that part must flow to reach the coolant. It is interesting to note that changing from cast-iron to aluminum pistons in this engine reduced the temper-

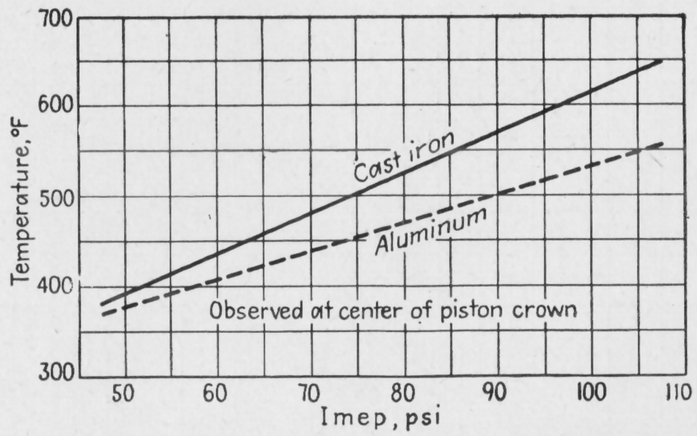

FIG. 172. Effect of power output on piston temperature with a constant cylinder-jacket water temperature. (*Underwood and Catlin, Trans. SAE, Vol.* 36, 1941.)

ature of the center of the piston crown to 550°F and that at the top-ring groove to 493°F.

Since liquid-cooled engines are normally operated at a constant value of coolant outlet temperature, the temperature of the combustion-chamber walls increases with power output as in Fig. 172. This follows from the fact that the temperature gradient is directly proportional to heat flow rate, which increases with power output. Air-cooled engines display the same characteristic, for the reference temperature is always at some point on the outside surface of the cylinder head or barrel. The magnitude of the effect of power output on combustion-chamber-wall temperature depends on the thermal conductivity of the metal and the length of the heat flow path.

Surface Heat-transfer Coefficient. As shown in a previous section, the rate of heat transfer from a gas to a solid surface of a given shape and form depends upon the mass flow rate of the gas passing over that surface. In a spark-ignition engine the mass flow rate of gases into, around in, and out of the cylinder is directly proportional to ihp. Thus, as a first approximation, the surface heat-transfer coef-

ficient should be expected to vary as the 0.80 power of the ihp [see Eq. (15)]. Experimental data have confirmed this characteristic.

Effect of RPM and Power Output on Heat Loss. In reexamining the heat-transfer equation

$$Q = hA\,\Delta t$$

with a view to the over-all effects of practical operating variables such as rpm, it can be seen from the preceding discussion that the area term remains constant in any given engine. Although the length of

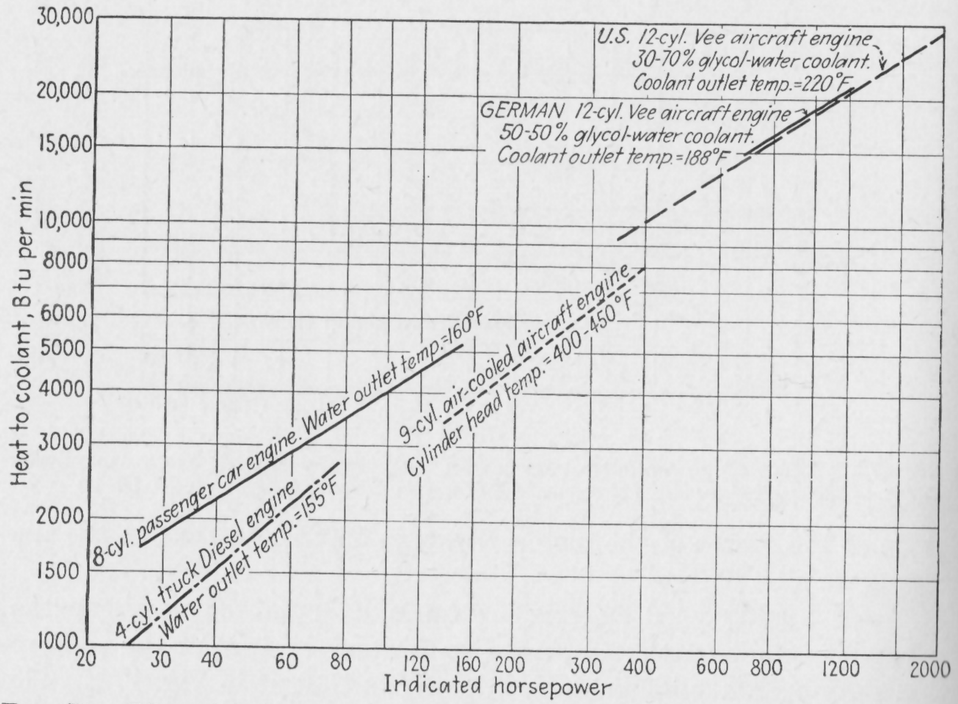

FIG. 173. Effect of ihp on the rate of heat loss to the coolant of representative engines (*Data from various laboratories.*)

time each element of area is exposed each cycle decreases with rpm, it is exposed more often so that the proportion of the time that a given area is exposed to hot gases is independent of rpm.[2] In a given spark-ignition engine the gas temperature remains substantially constant, while the combustion-chamber-wall temperature increases only a small amount with heat flow rate. This leaves the surface heat-transfer coefficient as the most important factor to be considered. Figure 173 gives data for the total amount of heat transferred (or rejected) to the coolant of a number of widely different engines. The agreement in the slope of the curves is good and confirms the argument given above that the heat loss should depend primarily on the

heat-transfer coefficient h and that the latter should vary with the mass flow of air through the engine cylinders. The ihp was used in Fig. 173, for it is directly proportional to engine air flow and is a more convenient parameter for engine-cooling work. The increase in combustion-chamber-wall temperature with power output at a constant coolant temperature causes the slope of the curves in Fig. 173 to be somewhat less than 0.80. In the spark-ignition engines for which curves are shown, the slope varies from 0.62 to 0.78. The heat loss Q in Btu per minute from an engine may be expressed by the formula

$$Q = K(\text{ihp})^n \tag{16}$$

where K and n are constants that depend on the design of the engine and the coolant temperature. The value of K commonly runs from 150 to 250, while that of n will run from 0.55 to 0.78. Since the data for each engine in Fig. 173 were taken for both high and low loads over a wide range of speeds, it is also evident from Fig. 173 that the heat rejection rate from the engine to the coolant is dependent only on ihp because all the points for each engine plot on a common straight line. It is true that piston-ring frictional losses cause small variations in the heat loss to the cylinder walls. These variations are responsible for some increase in heat loss with an increase in rpm at constant ihp, but such deviations are small, seldom exceeding 5 per cent.

The heat loss to the combustion-chamber walls of diesel engines varies in much the same manner as in spark-ignition engines in spite of fundamental differences in operating conditions. That is, the gas temperatures in diesel engines increase rapidly with power output at a given rpm while the air flow remains practically constant, whereas just the reverse is true in spark-ignition engines. The air flow in a diesel is, of course, approximately proportional to rpm. Thus the heat-transfer coefficient is primarily dependent on rpm, whereas the temperature difference between the combustion-chamber gases and the cylinder walls depends on the imep. Although the details of the relationships are somewhat complex, the combined effects of the various factors are such that the heat rejection to the coolant of diesel engines varies approximately as the 0.8 power of the ihp, or in essentially the same manner as in spark-ignition engines.[5]

Effect of Cylinder Temperature on Heat Loss. Increasing the operating temperature in any type of engine will reduce the heat loss, for it reduces the temperature difference between the hot gases and the metal in the cylinder. This is evident in Fig. 174, which shows

the effect of varying the coolant temperature on the heat loss from a liquid-cooled engine for each of three different types of coolant. Since the heat-transfer coefficient is higher for water than for ethylene glycol (see Fig. 169), the metal temperatures are lower with water and the heat rejection rate is higher at a given coolant temperature. The heat-transfer coefficient for transformer oil is about equal to that for ethylene glycol; hence the heat rejection rates for the two are very nearly the same.

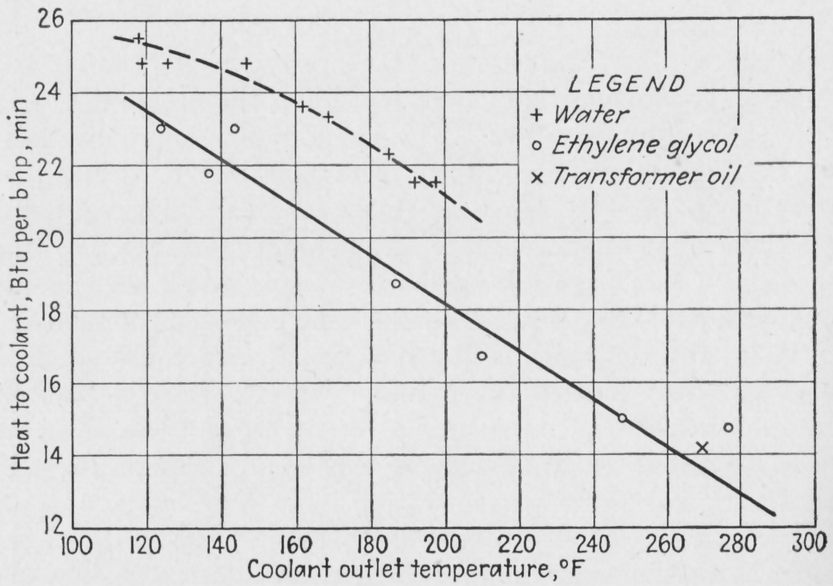

FIG. 174. Effect of coolant outlet temperature on the rate of heat loss to the coolant of a six-cylinder in-line liquid-cooled aircraft engine operating at 270 hp and 1,450 rpm. (*Weidinger, NACA Tech. Memo.* 649.)

Effect of Fuel-Air Ratio. The temperature of the burned gases in the combustion chamber varies with fuel-air ratio, the peak value being reached near the chemically correct mixture. Since spark-ignition engines normally operate near this point and since appreciably leaner fuel-air ratios are impracticable, the heat loss to the coolant is not very greatly affected by fuel-air ratio except under rich mixture conditions. The reduced gas temperatures under the latter conditions give a reduced heat rejection rate. This offers a means of keeping metal temperatures within allowable limits under critical cooling conditions. It is for this reason that large air-cooled aircraft engines employ very rich mixtures for high-power conditions. Since exhaust-valve cooling may be critical near peak load conditions in any type of spark-ignition engine, the fuel-air ratio is sometimes increased in

other than air-cooled aircraft engines in the high-power range to prevent valve burning.

Effect of Spark Timing. Advancing the spark increases the length of time during which the combustion-chamber walls are exposed to the hot burned gases. This results in an increase in the temperature of the combustion-chamber parts and an increase in the rate of

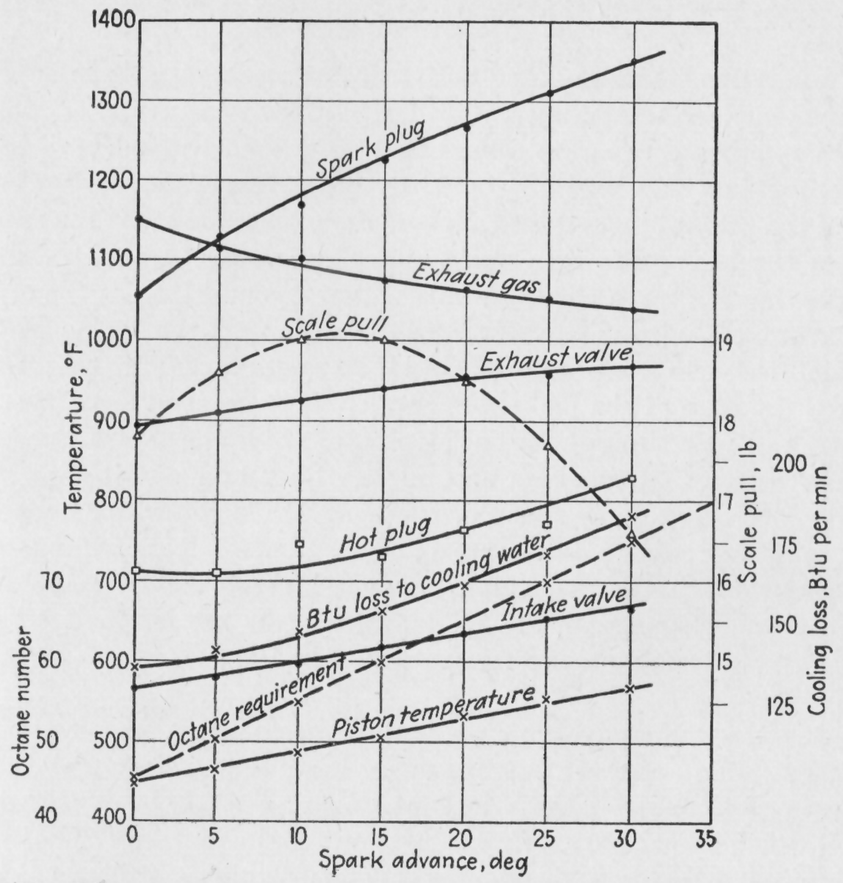

FIG. 175. Effect of spark advance on the temperature of combustion-chamber parts and on the heat loss to the coolant. (*MacCoull, Trans, SAE, Vol.* 34, 1939.)

heat rejection to the coolant, as indicated in Fig. 175. On the other hand, retarding the spark increases the exhaust temperature; for the amount of work done on the expansion stroke is reduced, and hence a greater amount of heat is left in the exhaust gas. These factors may cause an increase in exhaust-valve temperature if that part is not well cooled and may, under critical conditions, result in valve burning when the spark is retarded excessively.

Effect of Knocking Combustion. While knock does not increase the average temperature of the gases in the combustion chamber, it

does set up pressure waves that cause violent agitation of the gases in "pockets" of the combustion chamber. Although the total heat rejection from the engine is not greatly affected by knock, certain parts such as the spark plug, piston, or points in the cylinder head usually show a marked increase in temperature.

HEAT TRANSFER TO THE COOLANT AND COOLANT FLOW IN LIQUID-COOLED ENGINES

Evaporative Cooling. The heat-transfer mechanism is so dependent upon coolant temperature and flow rate that each type of coolant system requires a separate discussion. Many early models of internal-combustion engine were cooled by admitting water to the lower part of the coolant jackets and allowing it to boil off so that the greater part of the heat went into vaporizing the water. A tank of water was provided to serve as a reservoir. It was found that, if the piping were arranged to provide water circulation through the engine by convection, the evaporation rate could be reduced. The cool water entered the base of the jackets, expanded as it received heat from the engine, and rose through an outlet pipe to enter the top of the tank. The air convection currents around the tank and outlet pipe were sufficient to carry away much of the heat, while the water boiled off at a sufficient rate to take care of the balance. This arrangement also gave more uniform cooling by reducing the water temperature rise through the engine. Such a system can be designed to give very well controlled engine temperatures and is still in use for some applications.

Another evaporative cooling system has been adapted to both automobiles and aircraft but has never been very widely used. The water in the cooling jackets is kept under pressure by a throttling valve, which allows only steam to pass out of the jackets. The steam leaving the engine in such systems is condensed in a "radiator," from which the condensate is returned to the engine. This arrangement has the advantage of giving somewhat higher engine and heat-exchanger temperatures for a given pressure on the system but has the disadvantage that it requires a moderately complex set of control equipment.

The Thermosiphon System. Automotive engines must be provided with a cooling system that gives little or no coolant loss from evaporation, for otherwise relatively large amounts of water would be required. Many early automobiles made use of *thermosiphon* cooling systems, in which water entered the bottom of the cylinder block, moved upward by convection as it took on heat, and passed out of

the top of the cylinder head to the top of a radiator, or heat exchanger, having a large surface area exposed to colling air. The systems looked much like that in Fig. 176 except that no pump or thermostat was provided. The difference in density between the coolant in the radiator and that in the engine and outlet pipe was sufficient to set up circulation. The low circulating velocity could be employed because the principal heat-transfer mechanism in the engine jackets was boiling.

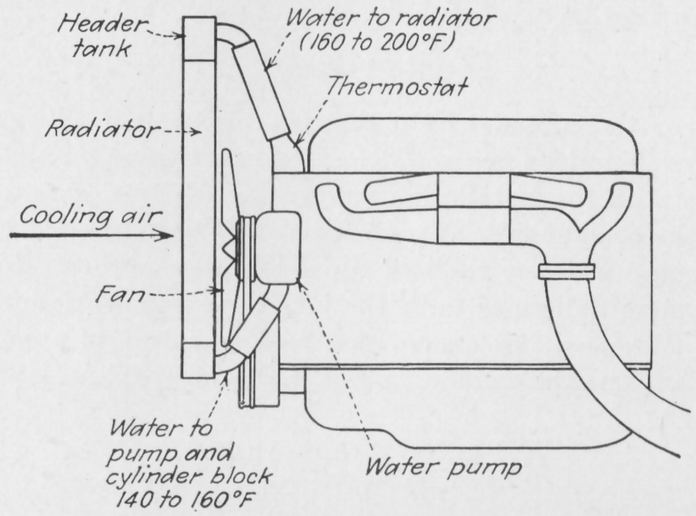

FIG. 176. Typical automobile-engine cooling system.

The thermosiphon system has the basic advantage of simplicity, but the temperature rise is inherently high and the radiator required relatively large because of the low circulation rate. Evaporation losses are likely to be high so that frequent filling is necessary, especially in hot weather. These conditions more than justify the addition of a water pump to increase the rate of circulation.

Forced-flow Liquid-coolant Systems. Practically all modern automobile engines employ cooling systems similar to that in Fig. 176. They make use of a centrifugal pump driven by the crankshaft. In pumps of this type the pressure rise is proportional to the square of the speed,

$$\Delta p_1 \cong N^2 \tag{17}$$

where Δp_1 = the pump pressure rise, psi
N = pump-impeller rpm

The pumps commonly used may be classified as being of one of two types, depending on the shape of the impeller vanes. The pressure rise through a pump having an impeller with radial vanes is prac-

tically independent of the rate of coolant flow through it, while one having an impeller with backward-sweeping vanes will give a continuously higher pressure rise as the flow is reduced.

Since the coolant flow through the engine and connecting lines is turbulent and since the pressure drop through the radiator tubes—where the flow is laminar—is a relatively small part of the total, the pressure drop through the system will vary nearly as the square of the flow. That is, the engine coolant passages act as a series of orifices. This may be expressed algebraically as

$$\Delta p_2 \cong (\text{gpm})^n \tag{18}$$

where Δp_2 is the pressure drop through the system and gpm is the coolant flow in gallons per minute. The exact value for the exponent n will vary with the installation but commonly runs around 1.8.

Since the coolant flow at each speed will stabilize at a value such that the pump pressure rise will equal the pressure drop through the coolant system and since both the latter vary approximately as the square of the speed, *the coolant flow is almost directly proportional to engine rpm.* That is, combining Eqs. (17) and (18) and letting $n = 2$ give

$$N^2 \cong \Delta p_1 \cong \Delta p_2 = (\text{gpm})^2$$

Hence

$$\text{gpm} = aN \tag{19}$$

where a is a constant for the pump and system in question. The above reasoning applies either to cooling systems with no thermostatically controlled valve designed to limit the coolant flow to give a constant coolant outlet temperature or to thermostatically controlled systems when the thermostat valve is wide open. Since the latter case represents the critical operating condition, Eq. (19) is of value in analyzing the performance of any forced-flow system.

Effect of Boiling in Forced-flow Cooling Systems. The most important consideration in cooling is the prevention of hot spots. In forced-flow cooling systems the heat transfer rate may be governed by coolant velocity if the coolant temperature is well below the boiling point, or the principal heat-transfer mechanism may be boiling if the coolant temperature is close to the boiling point. Figure 177 shows the temperature difference between the hot metal surface and the bulk temperature of the coolant for these two types of condition for a series of four constant heat transfer rates. The hyperbolic curves for the conventional heat transfer relation (where boiling is not a factor) were computed from Fig. 169, while the data for the

straight horizontal lines were taken from Fig. 170 for the case where the coolant is at the boiling temperature. A number of important points may be deduced at once from Fig. 177. If boiling is not a factor, the temperature of any given portion of the combustion-chamber

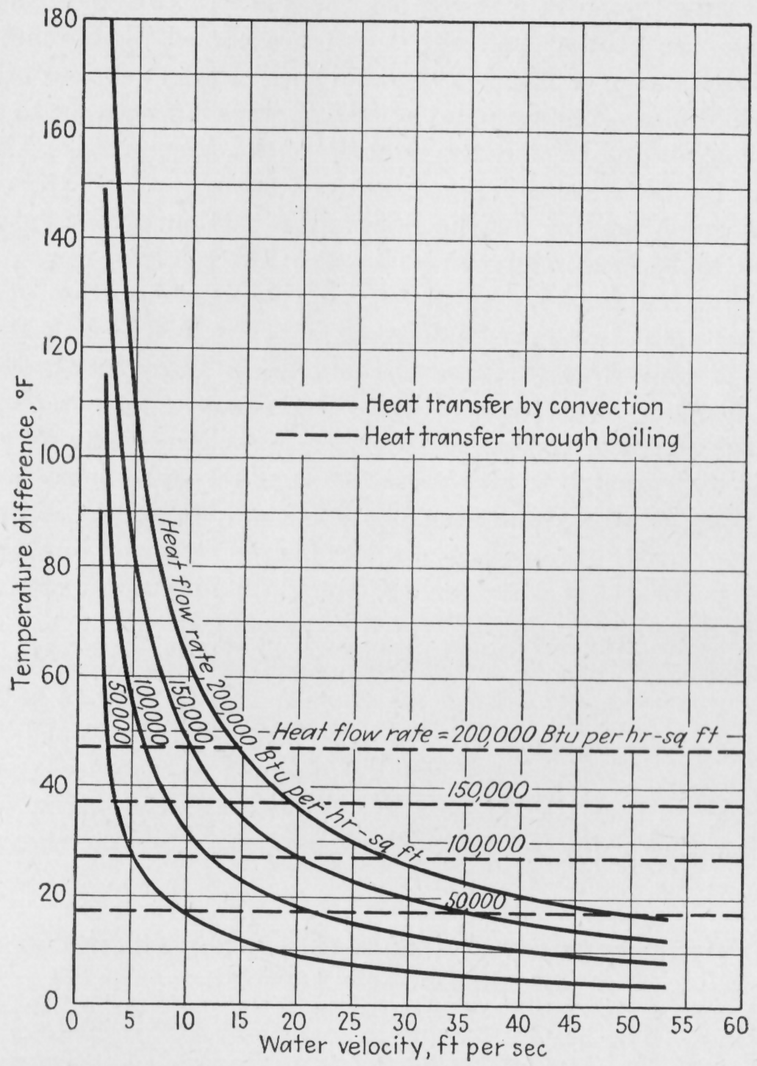

FIG. 177. Effect of cooling-water velocity on the temperature difference between the metal surface and the mean temperature of the water as computed from Figs. 169 and 170 for four constant heat-flow rates.

walls will depend on the coolant velocity over that portion. The complex cored coolant passages in a cylinder head make it difficult to secure uniform coolant velocities at all points. If they are not laid out just right, some regions will run much hotter than others. Serious thermal distortion and even hot spots may result. Again, if the

power output of an engine in which the cooling has been considered satisfactory is increased, the temperature difference between the hottest and coolest points in the cylinder head is often increased sufficiently to make the cooling unsatisfactory. Fortunately, with water-cooling systems operating at temperatures of 160 to 180°F, the spots over which the coolant velocity is low are cooled by boiling. The heavy dotted curve of Fig. 177 shows a case of this sort. If ethylene glycol, which has a boiling point of 387°F, were substituted as a coolant in such an engine and the coolant outlet temperature kept the same, the temperature difference between the metal and the coolant would be much greater for the nonboiling portion of the curve and would be multiplied many times in the low-velocity range, where water boiling had kept hot spots from developing.[6,7]

Another point that may be deduced from Fig. 176 is that, although the coolant flows from the base of the cylinder block up through the cylinder head in most engines, there is no reason why it could not flow in the opposite direction. In fact, for engines having coolant flow rates high enough to give pressure drops through the coolant passages of 5 or 10 psi, cylinder-barrel areas in which boiling is the principal heat-transfer mechanism would operate at lower temperatures with the reversed-flow arrangement, for the boiling point of the coolant would drop by 20 to 40°F.

The heat rejection to the coolant varies with power output, as pointed out previously. Since the coolant flow is fixed at any given rpm, the coolant temperature rise will increase with horsepower at any given rpm. Engines are usually designed to give a coolant flow high enough so that the temperature rise through the engine will not exceed about 20°F for the peak power condition with the thermostat, if present, wide open.

HEAT TRANSFER TO THE COOLANT AND COOLANT FLOW IN AIR-COOLED ENGINES

Air Flow. The low heat-transfer coefficient for air makes it necessary to increase both the surface area and the temperature differential between the metal and the coolant. While some small low-power engines depend upon convection currents to circulate cooling air, forced circulation is necessary for most engines. For such applications as aircraft and motorcycles the velocity of the vehicle is usually sufficient to induce an adequate cooling-air flow over the cylinders. For other applications, a cooling fan or blower is essential. The higher output engines also require a directed air flow to give uniform

cooling. This is necessary because the upstream side of a cylinder placed in an air stream will cool satisfactorily, but air will not flow in between the fins to cool the downstream side of the cylinder as well as the upstream side unless baffles such as those shown in Fig. 178 are employed. Since the air flow over baffled cylinders is turbulent, the flow rate varies as the square root of the cooling-air pressure drop, or *baffle pressure drop*, across the engine.[3,4,8] Thus the engine baffle

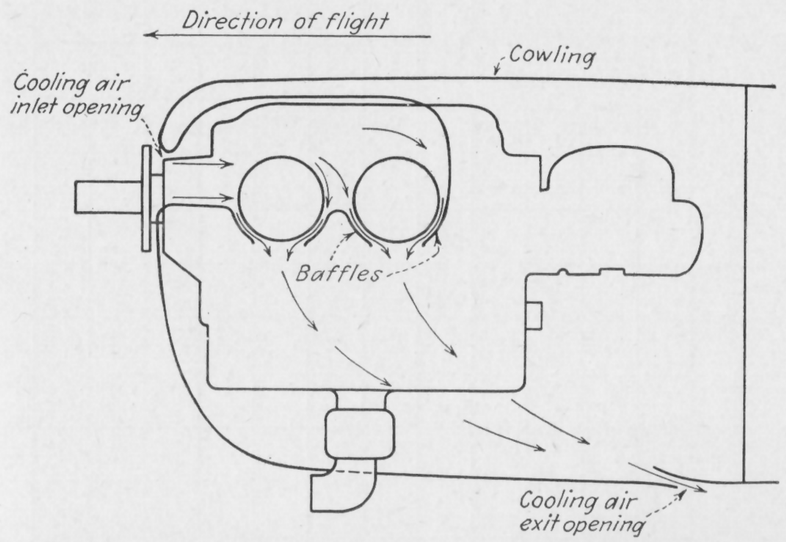

FIG. 178. Engine cowling and cylinder baffle arrangement used on a four-cylinder horizontally opposed light-plane engine.

passages may be considered as a group of orifices similar to the coolant passages of liquid-cooled engines. If a blower is used, the pressure that it is capable of developing will vary as the square of its speed. Thus the air flow over the cylinders of an engine fitted with an engine-driven fan or blower will be directly proportional to engine rpm in the same way as the liquid-coolant flow in a forced-flow system.

No cooling fan is provided in aircraft engines as a rule, for the air blast from the propeller will provide some cooling at low air speeds in the same manner as a fan. At air speeds above about 100 mph the principal factor setting up a cooling-air flow is the velocity pressure built up in front of the engine cowling in flight. Helicopters are, of course, an exception and require engines with blower cooling.

Heat Transfer from Finned Surfaces. The effects of both fin spacing and fin width on the rate at which heat will be transferred from a base surface to a cooling-air stream are quite important. Perhaps the most convenient method of expressing this characteristic is to give it in terms of total heat-transfer rate per square inch of the

surface from which the fins protrude rather than per square inch of area of the fins themselves.[9] Figure 179 shows the effect of fin width on the over-all heat-transfer coefficient for cylinders with three different fin spacings. In comparing the three curves of Fig. 179 it is evident that it is possible to have the fins too close together, although production problems are such that it is difficult even to approach the

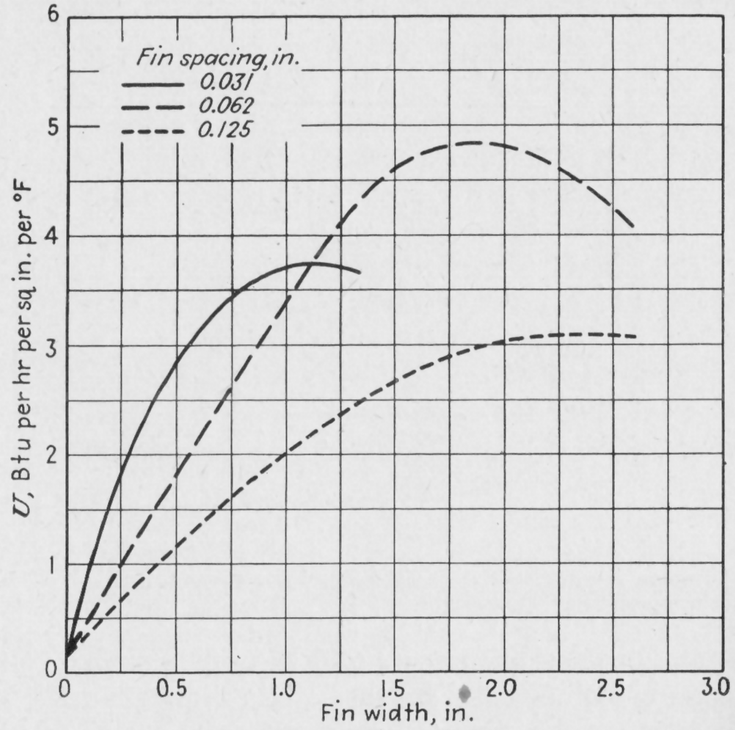

FIG. 179. Effect of cooling-fin width and spacing on the rate of heat transfer per square inch of cylinder-wall area for a representative cooling-air flow. (*Brevoort and Joyner, NACA Tech. Rep't.* 674, 1939.)

optimum spacing. Most air-cooled engines employ fin spacings of 4 to 12 fins per inch so that the lowest curve of Fig. 179 may be considered as representative of practicable fin spacings. The practicable fin width from the manufacturing standpoint varies with fin spacing from about two inches for the more widely spaced fins to less than an inch for closely spaced fins.

The heat-transfer coefficient as conventionally determined for finned cylinders varies with the cooling-air velocity at somewhat less than the 0.8 power indicated by Fig. 169, depending on both the fin width and the heat conductivity of the fin material.[10] This is to be expected; for the reference temperature is taken at the root of the fins, and the temperature drop between the root and the tip of the

fins increases with heat-flow rate so that the effective temperature difference between the fin and the cooling air is reduced. This factor may reduce the rate at which the heat-transfer coefficient varies with the cooling-air flow to as little as the 0.5 power of the latter.

Factors Determining Cylinder-head Temperatures. Air-cooled engines, unlike liquid-cooled engines, do not operate at any particular cylinder temperature. Instead, the cylinder-head temperature—the point commonly used as the reference engine temperature—is allowed to increase until equilibrium is reached for that power output and the prevailing cooling conditions. This works out rather well in practice, for only a relatively small change in cylinder-head temperature is required to take care of a relatively large change in power output. The reason is that, not only does an increase in cylinder-head temperature increase the temperature differential between the metal and the cooling air so that the heat rejection rate for a particular cooling-air flow is increased, but it also reduces the temperature differential between the metal and the hot gases in the combustion chamber. The latter reduces the amount of heat that has to be rejected to the cooling air in the same manner as in Fig. 174.

Since it is important from the standpoint of both detonation and lubrication that the cylinder-head temperature should not exceed about 500°F and since a serious power loss accompanies overcooling, the relation between the various factors concerned in air cooling is of particular interest. In test work the prime variables are head temperature, power output, fuel-air ratio, and the baffle pressure drop, which serves as a convenient measure of the cooling-air flow. Campbell[8] has shown that a close approximation is given by the empirical expression

$$t_h - t_a = \frac{K(\text{ihp})^{0.4}}{(\text{isfc})^{0.55}(\sigma\,\Delta p)^{0.27}} \tag{20}$$

where t_h = cylinder-head temperature, °F
t_a = cooling-air inlet temperature, °F
ihp = indicated horsepower
isfc = indicated specific fuel consumption, lb/(ihp)(hr)
σ = ratio of cooling-air density to that of standard air
Δp = baffle pressure drop, in. H_2O
K = a constant for each engine and baffle combination

The isfc term is used in place of fuel-air ratio. The equation applies only for mixtures richer than the best-economy value. That this equation is reasonable may be gathered from a somewhat different form obtained by squaring the terms and rearranging to give

$$(\sigma\,\Delta p)^{0.54}(t_h - t_a)^2 = \frac{K(\text{ihp})^{0.8}}{(\text{isfc})^{1.1}}$$

or

$$(\sigma\,\Delta p)^{0.54}(t_h - t_a) = \frac{K(\text{ihp})^{0.8}}{(\text{isfc})^{1.1}(t_h - t_a)}$$

Multiplying the baffle pressure drop by the air-density ratio provides for variations in air density so that the term $\sigma\,\Delta p$ represents air-weight flow, as may be shown by a simple derivation (see page 425). The product of the weight of cooling-air flow and the temperature difference between the engine and the air gives a value roughly proportional to the heat absorbed by the cooling air, and hence the left side of the above equation represents the rate at which heat is carried away from the engine. This must be equal to the rate at which heat passes into the cylinder heads. This, in turn, has already been shown to vary at a rate somewhat less than the 0.8 power of the ihp and to be approximately inversely proportional to the cylinder-head temperature in the normal operating range of temperatures. The heat rejection rate also varies inversely as the fuel-air ratio because of the effect of the latter on the gas temperatures. Figures 173, 174, and 46 indicate these relations. Thus it follows that Eq. (20) embodies the basic relationships developed earlier in this chapter.

Power Required for Air Cooling. The power required to force the cooling air through the baffle passages is of interest whether an engine-driven blower is employed or whether air is scooped from the air stream flowing past an aircraft engine. (The power in the latter case evidences itself in the form of drag on the airplane.) Since the product of the air density and the baffle pressure drop varies as the square of the cooling-air flow, we may write

$$\sigma\,\Delta p = K_1 W^2 \tag{21}$$

where W is the cooling-air-weight flow. The power required to supply the cooling air, called the *cooling horsepower* (chp), is

$$\text{chp} = \frac{\Delta p(\text{cooling-air flow, ft}^3/\text{sec})}{550} = K_2\,\Delta p\,\frac{W}{\sigma} \tag{22}$$

Substituting the value for W defined by Eq. (21),

$$\text{chp} = K_2\,\frac{\Delta p}{\sigma}\sqrt{\frac{\sigma\,\Delta p}{K_1}} = \frac{K_3(\sigma\,\Delta p)^{3/2}}{\sigma^2}$$

or

$$\sigma\,\Delta p = \left[\frac{(\text{chp})\sigma^2}{K_3}\right]^{2/3}$$

This may be substituted in Eq. (20) to put the latter in terms of chp instead of baffle pressure drop, as follows:

$$t_h - t_a = \frac{K(\text{ihp})^{0.4}}{(\text{isfc})^{0.55}}\left\{\left[\frac{\sigma^2(\text{chp})}{K_3}\right]^{2/3}\right\}^{-0.27}$$

Since the fuel-air ratio normally does not vary appreciably, the isfc

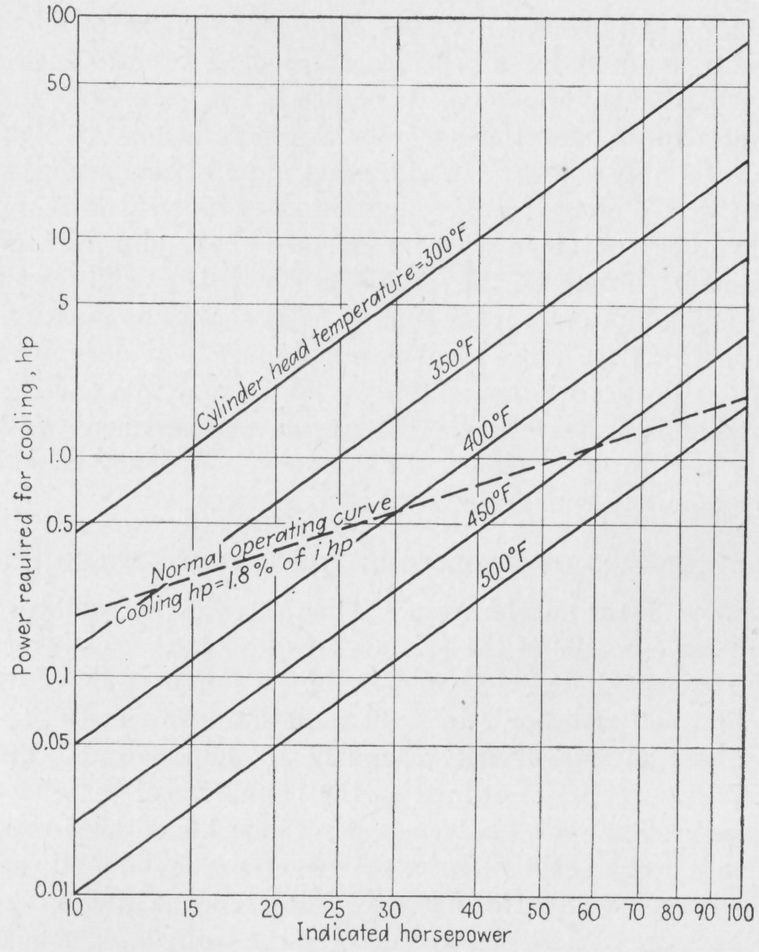

FIG. 180. Power required to force cooling air through the baffle passages of a four-cylinder horizontally opposed engine for various cylinder-head temperatures with 100°F cooling-air inlet temperature. Curves were based on data for a representative engine having a well-designed baffle system and extended by means of Eq. (23).

term can be included with the constant K and the equation simplified to give

$$t_h - t_a = K_4(\text{ihp})^{0.4}\left[\frac{\sigma^2(\text{chp})}{K_3}\right]^{-0.18}$$

Rearranging to give an explicit expression for chp,

$$\left[\frac{\sigma^2(\text{chp})}{K_3}\right]^{0.18} = \frac{K_4(\text{ihp})^{0.4}}{t_h - t_a}$$

$$\text{chp} = \frac{K_5(\text{ihp})^{2.2}}{\sigma^2(t_h - t_a)^{5.6}} \tag{23}$$

The significance of Eq. (23) is best seen from a set of curves constructed for a specific case. Figure 180 demonstrates the effect of ihp on the chp required for a representative light-aircraft engine fitted with a very good set of cooling-air baffles. The excessively high cooling power required for the lower head temperatures at high power outputs is forcibly shown. If wider and more closely spaced fins had been provided, a considerable reduction in cylinder-head temperature could have been obtained with no increase in the chp for a given set of conditions. Similarly, the same engine without baffles would have required much more power for cooling.[4] It should be mentioned that the chp in Eq. (22) is the ideal power required—it does not include the frictional losses or other sources of inefficiency in a cooling blower or aerodynamic losses outside the engine compartment if the slipstream is used in an airplane. (The over-all efficiency of cooling-air blowers commonly runs from 50 to 80 per cent.)

CHARACTERISTICS OF RADIATORS FOR LIQUID-COOLED ENGINES

Crossflow Heat Exchangers. The automobile radiator is the most familiar example of the various crossflow heat exchangers in use in the automotive industry. In spite of the connotations of its name, the automotive "radiator" does not radiate much heat to the cooling air—the heat is transferred primarily by conduction. The basic heat-transfer principles outlined at the beginning of the chapter and illustrated by Fig. 169 apply. Other examples of heat exchangers having similar characteristics are passenger-car heaters, oil coolers for trucks and tractors, aircraft oil coolers, intercoolers, and wing-deicing heat exchangers. Most of these involve the transfer of heat between a liquid and air or exhaust gas, although in some the heat passes from one gas to another or from one liquid to another. Water-cooled oil coolers for trucks are an example of the latter. The former constitute by far the larger group and present the more difficult design problems because of the low heat-transfer coefficients of gas films. Figure 181 shows the temperature distribution across the inlet and outlet faces for a liquid-to-air heat exchanger. Although the logarithmic mean temperature difference between the two fluid streams is the more general criterion of the heat transfer rate, the difference between the inlet temperatures of the two streams is the more convenient working parameter and hence is widely used. Since the heat transfer rate depends upon the heat-transfer coefficients, it will vary with the rate of flow in either fluid stream. Since the liquid flow rate is often fixed and since the cooling-air temperature rise is of little practical importance because the air is simply discharged into the vehicle slip stream, the liquid temperature drop per degree inlet-temperature difference —sometimes referred to as the "cooling effectiveness"—is often used in place of the actual heat-dissipation rate in Btu per minute.

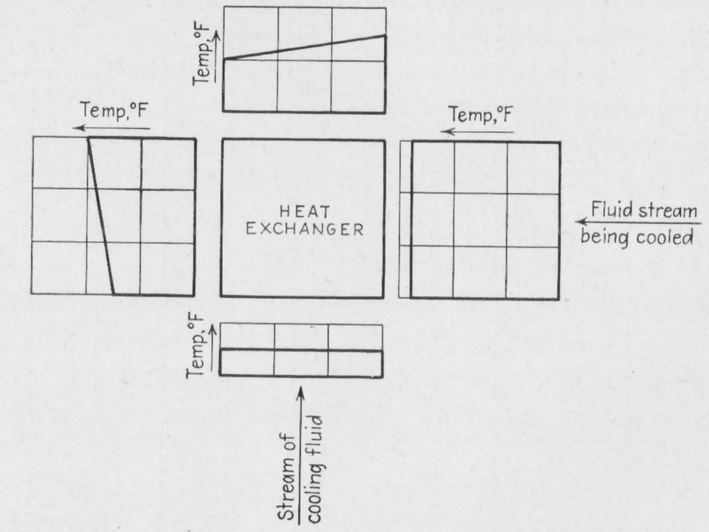

FIG. 181. Temperature distribution in the fluid streams entering and leaving a crossflow heat exchanger.

Radiator Construction. The low heat-transfer coefficient for air as compared with water makes the use of extended surfaces on the air side of the radiator core very much worth while in that it reduces both the weight of the core itself and the weight of coolant required to fill the core—an item of particular importance when antifreeze solutions are required. A widely used radiator-core construction is the fin-and-tube type, the most common form of which consists of thin, flat tubes passed through a stack of closely spaced fins to provide a large amount of heat-transfer surface in contact with the cooling air. All contact points between the fins and tubes are soldered to give a minimum resistance to heat flow. A second type of construction employs thin-walled circular copper tubes about ¼ in. in diameter, with the ends expanded to give a somewhat larger hexagon. These tubes are nested to form a "cartridge-type" core and the tube ends dipped in

solder to join them together. The circular form of the tubes makes this construction capable of withstanding relatively high pressures so that it is well suited to certain applications. Oil coolers for aircraft engines are made in this form partly because high pressures may be

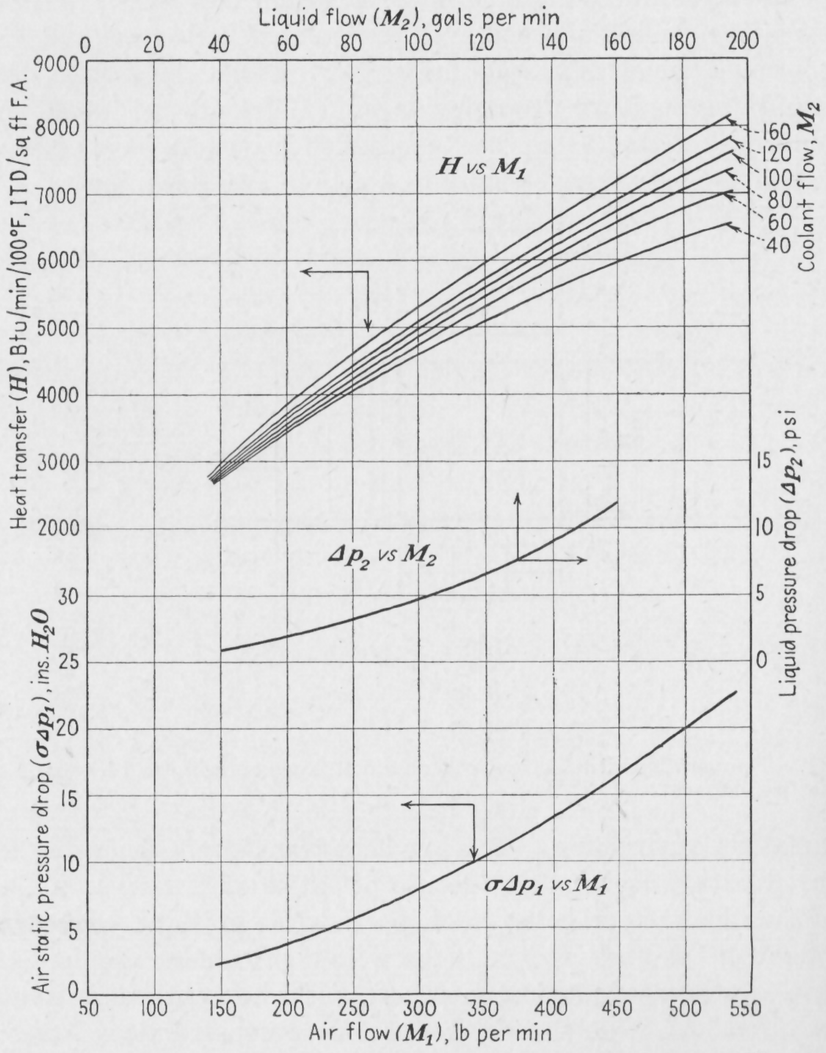

FIG. 182. Performance curves for a liquid-to-air radiator core section. The quantity H is the cooling effectiveness. (*Harrison Radiator Div., General Motors Corp.*)

developed in the cooler when the engine is started in cold weather with very viscous oil. However, since the area in contact with the cooling air flowing through the tubes is almost identical with that of the liquid passing between the tubes, this construction is heavier for a given heat-transfer capacity than some other types. A third type of construction employs shapes of formed thin sheets assembled so

that a honeycomb structure results. Water flows downward in the passage formed between alternate rows of hexagons. Several modifications of this type have been made to increase the proportion of the area exposed to the cooling air.

Radiator Performance. The performance characteristics of a radiator are often shown by a set of curves similar to that in Fig. 182.[11] The effects of changes in both air flow and coolant flow on the heat-dissipation rate are shown by the curves for a series of constant coolant flows. The additional curves for the air and coolant pressure drops through the radiator as a function of the respective flow rates are also essential, for a satisfactory heat-transfer rate must be obtained at the cooling-air flow which will result with the radiator installed in the vehicle. At the same time, the coolant pressure drop through the radiator determines the coolant flow for a given engine–coolant-pump–radiator combination. Note that the coolant flow rate has relatively little effect as compared with that of the air flow.

PROBLEMS IN THE DESIGN AND MAINTENANCE OF LIQUID-COOLANT SYSTEMS

Optimum Operating Temperature. A considerable amount of work has been carried out to determine the optimum operating temperature for engine cylinders. The principal considerations are wear, specific fuel consumption, full-throttle power output, and the radiator requirements.

Perhaps the most important consideration is cylinder-barrel and piston-ring wear. Figure 183 shows the rapid increase in wear rate as the jacket-water temperature is decreased below about 150°F. Similar investigations by many organizations have shown essentially the same characteristic. A high wear rate seems to be caused by condensation of moisture from the exhaust gases on the cylinder walls when the cylinder-wall-surface temperature drops below 212°F. Not only does the condensed water tend to corrode the cylinder wall and piston rings, but it also reacts with the oil to form sludge, which fouls the rings. As indicated in Fig. 171, the temperature difference between the cylinder-wall-bearing surface and the jacket water runs 50°F or more, depending on the engine design and the power output. As a result, jacket-water temperatures as low as 140°F may be employed without water condensation on the cylinder wall, with its attendant ill effects. Somewhat higher temperatures are desirable, however, both to provide a safe margin and because water condensation in the crankcase may otherwise occur and set up a general oil-sludging condition.

The effects of coolant temperature on engine power output and fuel consumption are interrelated. The largest single source of engine friction lies in the piston rings. The lower oil viscosities at the higher coolant temperatures often tend to reduce this factor sufficiently to give a noticeable increase in full-throttle power output for jacket

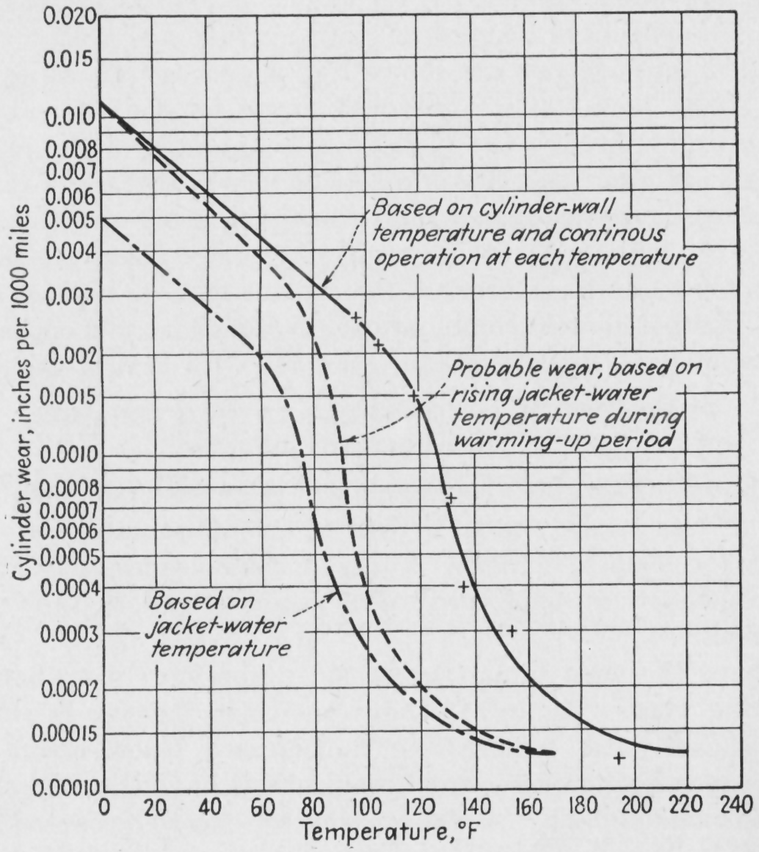

FIG. 183. Effect of temperature on cylinder wear (*Blackwood, Trans, SAE, Vol. 36,* (1935.)

temperatures up to 200 or 250°F. Higher coolant temperatures tend to reduce the density of the fresh charge flowing from the intake manifold into the cylinder sufficiently to more than offset any further reduction in friction for higher coolant temperatures. Coolant temperature also affects the full-throttle power output by affecting the knock or preignition limit. This effect is not large for most American motor fuels until metal temperatures in excess of about 500°F are reached. Higher metal temperatures may result in a pronounced drop in the detonation limit. The minimum bsfc obtainable is improved by increasing coolant temperatures both because of the

reduction in frictional losses and because of improved fuel vaporization at the time of ignition. The magnitude of this effect varies widely with the engine and the charge temperature in the intake manifold.

The radiator size required falls off very rapidly as engine coolant-outlet temperature is increased. Further, the amount of cooling air and hence the power required by the fan to force the air through the radiator falls off rapidly as the coolant-outlet temperature is increased in much the same way as the cooling power for an air-cooled engine was shown to fall off with increasing cylinder-head temperature in Fig. 180.

From the above discussion it appears that the engine coolant-outlet temperature should be at least 140°F and preferably higher to give a low rate of cylinder and piston-ring wear. Jacket-water temperatures of 200 to 250°F appear to be optimum from the standpoint of power output and fuel consumption, while, from the standpoint of the radiator, the higher the coolant temperature the better.

The boiling point of the coolant used has proved to be the decisive factor in establishing the coolant-system operating temperature. The low cost and excellent thermal characteristics of water have made it the principal engine coolant. Ethylene glycol has been the only other engine coolant to be used to any extent without dilution by water. Its poor heat-transfer coefficient and high cost as compared with water have limited its practical use to a relatively small number of specialized engines. Of course, the relatively high freezing point of water has made it necessary to add an antifreeze for winter operation, but the resulting solutions have properties very similar to those of water. Kerosene and other petroleum fractions have been employed as coolants, but the fire hazard has been too great for general use. Although both the alcohols and ethylene glycol are combustible, their water solutions employed as antifreezes are not.

Since most automotive equipment is designed to operate with water as a coolant, allowances must be made for a reduction in boiling point if alcohol is used as an antifreeze or if the vehicle is driven over mountains at altitudes of 5,000 or 10,000 ft, where the boiling point of water is reduced by 10 or 20°F. Allowance should also be made for the absorption of heat after the engine has been run for a time and then abruptly stopped. Unless the coolant temperature is appreciably below its boiling point before the engine is stopped, the latent heat given up by the hot metal-combustion-chamber parts will cause serious evaporation losses. With all these considerations in

mind it is not difficult to see why automobiles have been equipped with coolant systems designed to operate at between 140 and 180°F.

Coolant-temperature Control. The preceding section has shown the need for the proper control of coolant temperature. The blocking type of thermostat has been found to give the simplest type of automatic control. The most satisfactory arrangement is a thermostat located in the cylinder head at the coolant outlet. It can be set to start to open at, say, 145°F and continue to open progressively to the "wide-open" position at 160°F. Figure 184 shows the results of

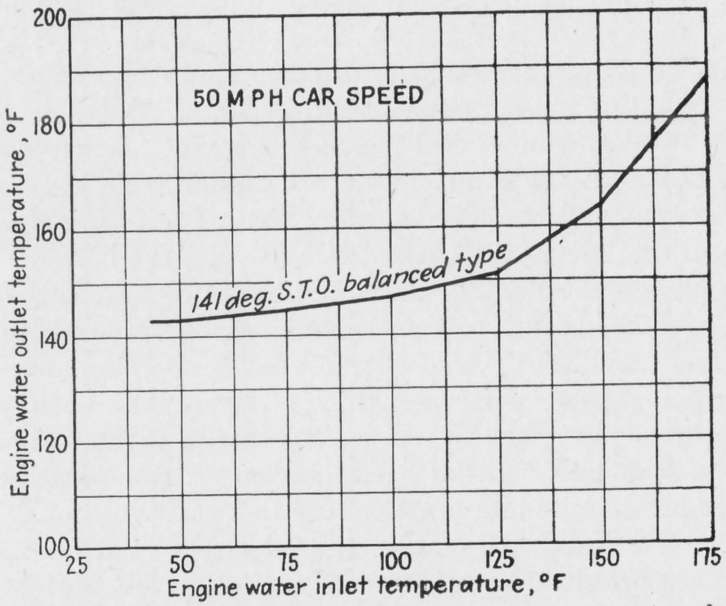

FIG. 184. Effect of water-inlet temperature on water-outlet temperature for a passenger-car engine with a blocking-type thermostat. (*Saunders, Trans, SAE, Vol.* 39, 1936.)

a full-scale engine test run with an installation of this type. It may be considered as showing the effect of the radiator cooling-air temperature upon both the inlet and outlet temperature of the coolant supplied to the engine for a car speed of 50 mph on level road. As the cooling air, and hence the coolant-inlet temperature, is increased, the coolant outlet temperature is held constant at 141°F by the thermostat until the radiator is unable to maintain a sufficiently low engine coolant inlet temperature with the small coolant flow permitted by the thermostat. Further increases in cooling-air temperature give progressively higher coolant-system temperatures. Above a coolant outlet temperature of 160°F the thermostat is wide open, the full coolant flow rate is obtained, and the coolant temperature is determined by the radiator cooling-air temperature and flow. It is general practice to select a radiator that will keep the coolant below the boiling point with 100°F cooling air and with the heat rejection rate given by the engine operating at full throttle at any normal operating road speed.

The cubic relation between fan horsepower and engine speed results in the fan absorbing as much as 10 per cent of the engine output at high speeds. This is much more than would be needed normally were it not for the fact that low-speed full-throttle cooling requirements

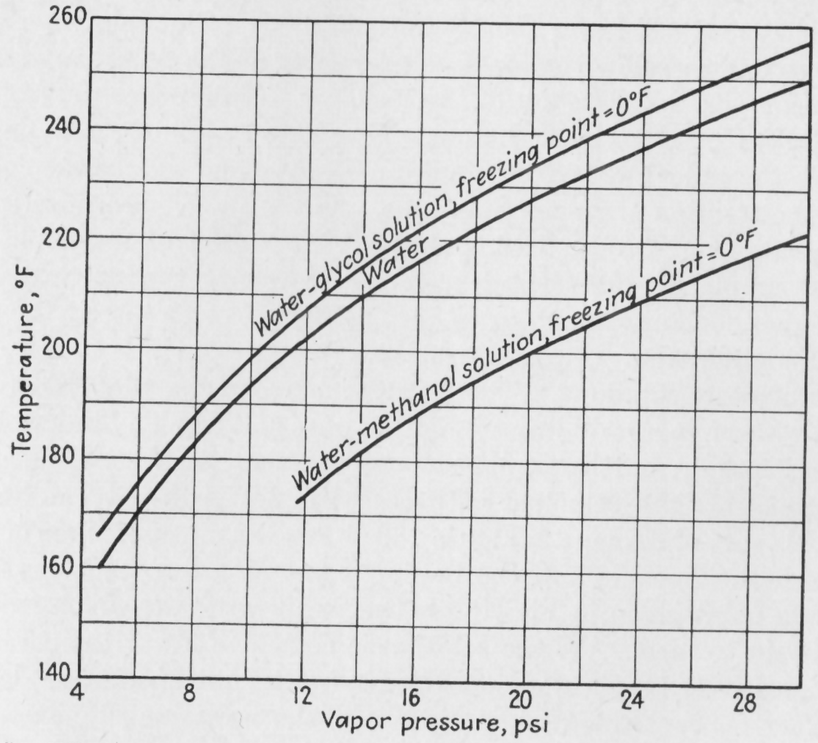

FIG. 185. Effect of pressure and of two antifreeze solutions on the boiling point of water.

fix the fan-blade pitch. This suggests that controlling the fan would be a more efficient way of regulating engine temperature than a blocking-type thermostat in the coolant system. Several devices of this sort have been developed, although none is yet in large-scale production at the time of writing. Some of them depend on control of fan-blade pitch, while at least one other depends on control of fan speed through a variable slip-type coupling on the fan drive shaft.

Pressure Cooling. Figure 185 shows the manner in which the boiling points of both water and antifreeze solutions vary with pressure and suggests a convenient means for increasing the allowable coolant operating temperature without abandoning water as a coolant.

Small changes in design may be required to permit pressurization, and an additional complication in the form of a pressure relief valve must be added. The latter prevents the escape of vapor from the system and allows the coolant to build up a pressure when boiling tends to occur. Unless care is taken in locating the coolant pump and in designing the passages between it and the radiator, cavitation may occur at the inlet to the impeller. That is, the pressure drop through the pump inlet passages together with the high coolant velocities and consequently low static pressures at the inlet edges of the impeller vanes will lower the pressure at the impeller inlet to a value of 1.5 to 6.0 psi below the pressure prevailing at the relief valve in the expansion space at the top of the radiator. This pressure reduction may be sufficient to cause boiling. The rate of boiling may be high enough to render the coolant pump ineffective and reduce the coolant flow to a fraction of its normal value. While these conditions can be taken care of by proper design, they are more difficult to handle than in the simple nonpressurized system. However, the advantages of the higher coolant temperatures obtainable will often justify the additional complications of pressurization.

Antifreeze Solutions. The 9 per cent expansion that takes place when water freezes will burst radiator tubes and crack cylinder blocks. Methyl, ethyl, and isopropyl alcohol together with ethylene glycol (prestone) are the principal antifreezes employed to prevent freezing and this sort of damage. Figure 186 shows the effect of three of these on both the freezing and the boiling point of water solutions. The freezing point given in Fig. 186 is the temperature at which ice crystals begin to form. As the solution is cooled to lower temperatures, a progressively larger amount of water freezes out of solution, leaving a higher concentration of antifreeze in the remaining liquid. As a result, solutions containing more than 10 per cent of alcohol or glycol will not freeze solid but will form a slush.[12] While this slush will not crack cylinder blocks or radiator cores, it will clog the small radiator passages sufficiently to prevent coolant circulation and hence is likely to cause local overheating in the engine when it is started from the cold condition. Even if this does not happen, with alcohol solutions most of the antifreeze may be boiled off. If the loss is not noticed, the remaining water may freeze and damage the engine after the next stop. "Slushing" of the radiator is likely to occur if the coolant temperature drops 5°F or more below the "freezing point" given in Fig. 186 for the solution.

Corrosion. Corrosion is always a problem in liquid-coolant systems.[12] The principal mechanism seems to be electrolytic action

either between dissimilar metals such as copper and solder in the radiator or between two different areas of the same material which are at different temperatures so that in effect a "thermocouple" is formed between the metal and the coolant. A particularly troublesome cause of corrosion is *aeration* of the coolant. Coolant splashing into a partly filled radiator will entrain considerable quantities of air in the form of

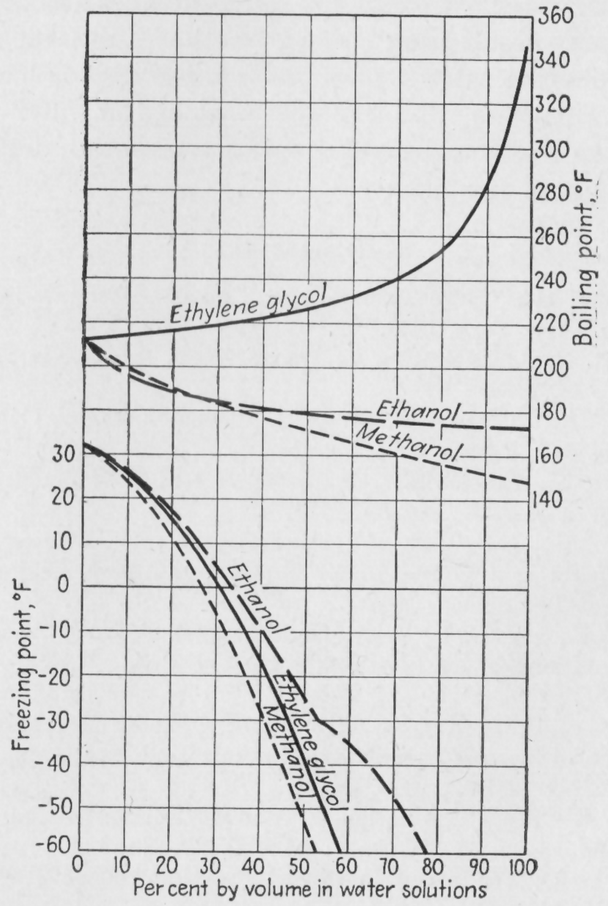

FIG. 186. Effect of antifreeze concentration on the boiling and freezing points of water solutions of common antifreezes.

small bubbles, which will remain in suspension and circulate with the coolant because of the turbulent-flow conditions that prevail. Air may also be drawn into the system on the suction side of the coolant pump either through slight leaks in the hose connections or through the seal on the pump impeller shaft. Oxygen will dissolve into the coolant from these air bubbles and will rapidly attack cast-iron, steel, and copper surfaces. The best and only effective way to prevent corrosion from coolant aeration is to prevent aeration by keeping the system full of coolant and free of air leaks.

Electrolytic corrosion can be almost eliminated by the use of inhibitors. Since it is especially likely to be troublesome when an antifreeze is used, the commercial products usually include an inhibitor so that no further chemical addition is necessary. Since the products of corrosion are rust, sludge, and scale, the problems of corrosion are closely associated with those of scale formation from mineral salts dissolved in the water. Clogging of radiator tubes and of engine coolant passages may result from either or both. The large temperature drop across adherent scale deposits may cause serious local overheating. To prevent these troubles, treatment of the water used for the radiator is often worth while and sometimes essential, depending upon the water supply of the locality.

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Problems

1. Given an engine in which the mean effective gas temperature runs 1200°F and cooling data showing that the heat rejection from each cylinder runs 2000 Btu/min with a cylinder-head-metal temperature of 300°F, estimate the heat rejection rate for a cylinder-head-metal temperature of 500°F.

2. Estimate the heat rejection in Btu per minute from a 225-hp liquid-cooled truck engine at full load by using an empirical percentage of the engine output.

3. If a racing-car engine were to be supercharged to give an intake manifold pressure of 36 in. Hg abs as compared with a normal full-throttle value of 27 in. Hg, what percentage increase in heat rejection to the coolant should be expected? If engine cooling had been barely adequate in the racing car for the unsupercharged engine, what increase in radiator size would be necessary for the supercharged model? Assume no pressurization in both cases.

4. If an increase in radiator size were impractical in the vehicle of Prob. 3, what pressure would have to be carried in the radiator to give adequate cooling with the original radiator?

5. What does the full-throttle coolant temperature rise normally run in liquid-cooled engines?

6. Indicate briefly the effect of each of the following factors on the coolant flow rate in a liquid-cooled engine: (*a*) rpm; (*b*) load; (*c*) coolant temperature.

7. To what extent would coolant temperature affect the temperature of a hot spot adjacent to an exhaust-valve seat at high engine outputs? What effects would coolant-system pressure have on the hot spot?

8. Indicate briefly the effect of each of the following on coolant-system corrosion: (*a*) aeration, (*b*) antifreezes such as alcohol or glycol, and (*c*) inhibitors.

9. Why is the coolant pump always placed on the coolant-inlet rather than the coolant-outlet side of the engine?

10. List three undesirable features of overcooling an engine. List three reasons for cooling an engine more than the minimum amount allowable without part damage.

CHAPTER XII

SUPERCHARGING

Mention has been made many times in previous chapters of the dependence of engine power output on the amount of air that the engine can pump through its cylinders per unit of time. One relatively easy way to increase the rate at which an engine will handle its working fluid is to *supercharge* it, *i.e.*, increase the pressure and density of the air in the intake manifold. Thus the principal object of supercharging is an increase in the power capacity of the engine. The potential gain in output must be weighed against the supercharger power requirements and the increased cost, weight, and complication of the supercharger. It has been found in practice that supercharging is worth while for heavy automotive equipment, the medium and large aircraft engines, and many of the larger diesel-engine applications such as railroad locomotive, stationary, and marine units.

In the larger engines it has been found possible to drive the supercharger by means of a turbine designed to salvage some of the energy in the exhaust gas and thus make possible a small gain in fuel economy. The performance characteristics and possibilities of such an arrangement are quite complex and will be considered in a later chapter. The material in this chapter will be confined to the problems associated with air compression for supercharging purposes. Since many of the compressors employed in gas turbines are of the same type and have the same operating characteristics as compressors used for supercharging, most of the material in this chapter applies equally well to both piston and turbine types of power plant.

RELATIONS BETWEEN THE CHARACTERISTICS OF IDEAL AND ACTUAL COMPRESSORS

An isentropic compression gives a lower temperature rise and requires less power than the actual process for any type of compressor likely to be used as an engine supercharger. While large low-speed well-cooled reciprocating air compressors may approach an isothermal compression, they are much too bulky and heavy for engine-supercharging purposes. Since an isentropic compression represents the ideal process for engine superchargers, one of the best and most

widely used ways of presenting the performance of an actual unit is in terms of the degree to which it approaches an ideal isentropic compression.

Pressure Ratio. Inasmuch as the delivery pressure given by a compressor of any type commonly used for supercharging varies directly with the inlet pressure, the pressure ratio has been found to be a very useful parameter. This quantity defines the delivery pressure independently of the inlet pressure at any given speed and flow rate. While a general proof of such a statement is difficult, it can be demonstrated readily for any type of unit currently in use. For example, its truth is quite evident in connection with a constant-displacement machine such as a reciprocating-piston and cylinder type of compressor in which the ratio of the final volume to the initial volume is fixed. Thus, for an isentropic compression,

$$\text{Pressure ratio} = \frac{P_2}{P_1} = \left(\frac{V_1}{V_2}\right)^k$$

The efficiencies of rotary-type compressors such as centrifugal and Roots blowers depend to a large extent on back leakage, which, in turn, depends on pressure ratio. Because of this, even in a machine such as a Roots blower, changing the inlet pressure will have little effect on curves of efficiency plotted against pressure ratio.

Power Input. The work ideally required for isentropic compression and delivery of a gas was derived on page 51, where it was shown that

$$\text{Net work for compression and delivery} = \frac{k}{k-1}(P_2V_2 - P_1V_1)$$

Since it is usually more convenient to work in terms of the pressure ratio P_2/P_1, the air inlet temperature, and the air-weight flow, a different form of the above relation is more useful. Such a form may be obtained by using the relationship $PV = WRT$.

$$\text{Net work} = \frac{k}{k-1}(WRT_2 - WRT_1)$$

But, for an isentropic compression, $T_2 = T_1(P_2/P_1)^{\frac{k-1}{k}}$; hence

$$\text{Net work} = \left(\frac{k}{k-1}\right)WRT_1\left[\left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}} - 1\right]$$

The value of $(k-1)/k$ is 0.283 for air in the temperature range of interest in supercharging work. The quantity within the brackets is

often referred to as the *Y factor*, values for which are given in Table III of the Appendix. On this basis the above equation reduces to

$$\text{Net work} = \frac{WRT_1Y}{0.283} = 0.1883WT_1Y \tag{24}$$

Adiabatic Shaft Efficiency. The efficiency of an actual compressor may be expressed as the ratio of the isentropic power input to the actual power input for compression between a given set of pressure limits. Thus the ratio of the isentropic work to the actual power input to the compressor shaft has been widely used and is known as the *adiabatic shaft efficiency*, η_s. If W is in pounds per second, then

$$\eta_s = \frac{550\,\dfrac{k}{k-1}\,WRT_1\left[\left(\dfrac{P_2}{P_1}\right)^{\frac{k-1}{k}} - 1\right]}{\text{compressor shp}} \tag{25}$$

Adiabatic Efficiency. Since it is usually difficult to measure the power input to a compressor, another parameter is more widely used than that defined above. One important basis for its use is that whatever energy goes into air-pumping losses remains in the air in the form of heat so that the air leaves the compressor at a higher temperature than would be the case for a reversible adiabatic compression. From the law of the conservation of energy, if there is no heat transfer, the flow of heat energy represented by this temperature increment must be equal to the difference between the isentropic and the actual power inputs. Thus the ratio of the isentropic to the actual air temperature rise across the compressor should give a value similar to the adiabatic shaft efficiency except for the fact that it would not include the effects of mechanical frictional losses in the bearings or heat transfer to the housings from the air being pumped. The adiabatic efficiency therefore has been defined as

$$\eta_{ad} = \frac{\text{isentropic temp. rise}}{\text{actual temp. rise}} = \frac{T_2[(P_2/P_1)^{\frac{k-1}{k}} - 1]}{T_2 - T_1} = \frac{T_1Y}{T_2 - T_1} \tag{26}$$

Volumetric Efficiency. If the compressor is of the constant-displacement type (such as a Roots blower), its air-delivery characteristics are often conveniently expressed in terms of the volumetric efficiency. As might be expected, this is defined as the ratio of the actual quantity of air pumped to the ideal quantity that would be induced by the compressor on the basis of the displacement of its pumping elements.

TYPES OF SUPERCHARGER AND THEIR CHARACTERISTICS

The great majority of the superchargers in use at the time of writing are of either the centrifugal or the Roots-blower type. The Lysholm compressor is an interesting third type the development of which began about 1934. Since its future in internal-combustion-engine work appears promising and since its characteristics are typical of an important class of compressors, it is also included in this chapter.

Roots Blowers. Roots blowers are widely used wherever pressure ratios in the range from 1.1 to 2.0 are desired.[1–3] An end view of a conventional unit is shown in Fig. 25. Air is trapped between the rotor lobes at the inlet and carried through the space between the rotors and the housings to the outlet port. Since the rotors are contoured so that they mesh together and close off the space between them as they rotate, air back leakage can take place only through the clearances. The amount of back leakage is minimized by reducing the clearances between the rotors and the housings to as small a value as possible. Deflections under load together with the effects of thermal expansion render it impractical to use clearances of less than 0.005 to 0.010 in., depending on the size of the unit.

Operation of Roots blowers is characterized by a throbbing noise caused by air rushing into each rotor space as it begins to register with the outlet port. The amount of noise has been reduced by employing three-lobed rotors instead of two-lobed rotors, machining a relief in the housing to give a more gradual opening of the exhaust port, and in some cases using helical instead of straight lobes on the rotors.[2]

Roots blowers are not well suited to applications requiring pressure ratios in excess of 2:1, for their efficiency falls off seriously at higher values. The reason for this may be seen most easily by examining the Roots blower in Fig. 25. The air in each rotor space is not compressed until the exhaust port is opened, at which point the pressure rises to the full delivery value. The rotor must then force out, not only the air it had brought in as a fresh charge, but the backflow air that rushed in to do the work of compression. The loss entailed by this factor is indicated by the crosshatched area in Fig. 187, which shows an indicator diagram for the compression process. If sketches are made for a series of diagrams similar to Fig. 187 but for progressively higher pressure ratios, it can be seen that the loss entailed by backflow compression becomes an increasingly important factor as the pressure ratio is increased.

A set of performance characteristics of a Roots blower are shown

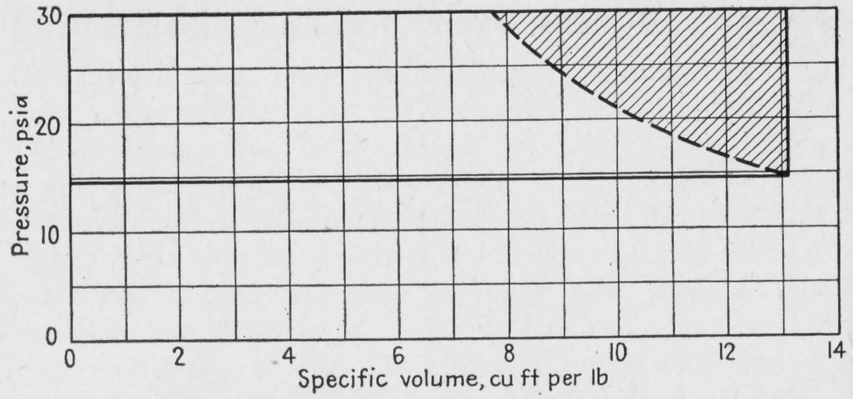

FIG. 187. Pressure-volume diagram for an idealized backflow compression from atmospheric conditions. The crosshatched area represents the extra work required by backflow compression as compared to an isentropic compression.

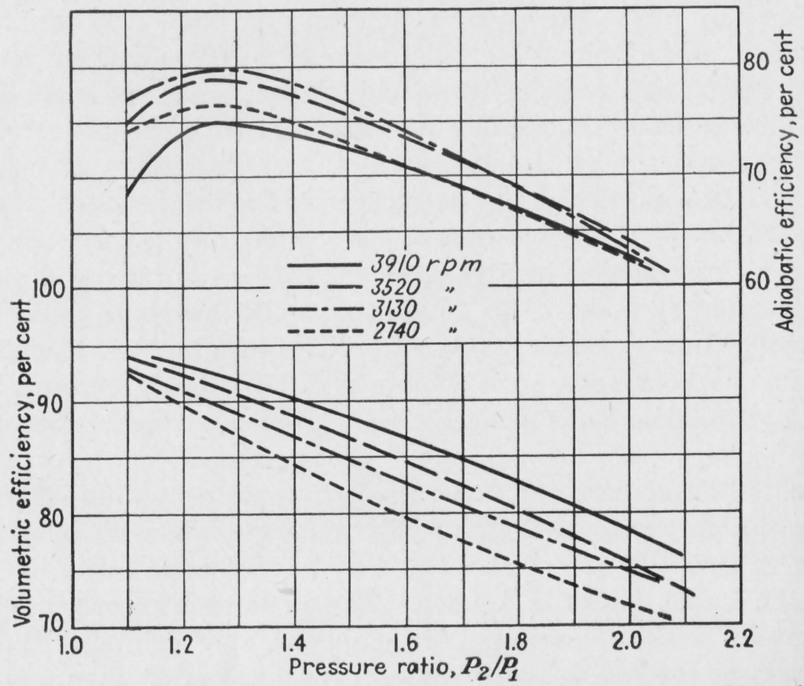

FIG. 188. Effect of pressure ratio on the volumetric and adiabatic efficiencies of a Roots blower for an automotive engine.

in Fig. 188. Note that the volumetric efficiency falls off rapidly at an almost linear rate with an increase in pressure ratio, partly because of the compressibility of the air and partly because of the increased back-leakage rate. The adiabatic efficiency of the unit is also given in Fig. 188. That parameter, too, falls off with pressure ratio, in this case because of both increased back leakage and the proportionally greater difference between isentropic and backflow compression

(as indicated by the crosshatched area of the indicator diagram in Fig. 187).

All the more important characteristics of a compressor may be presented in the form of a single chart, or "map" (see Fig. 189), in which pressure ratio and flow rate are used as coordinates and curves are given for a series of constant speeds. Lines of constant adiabatic

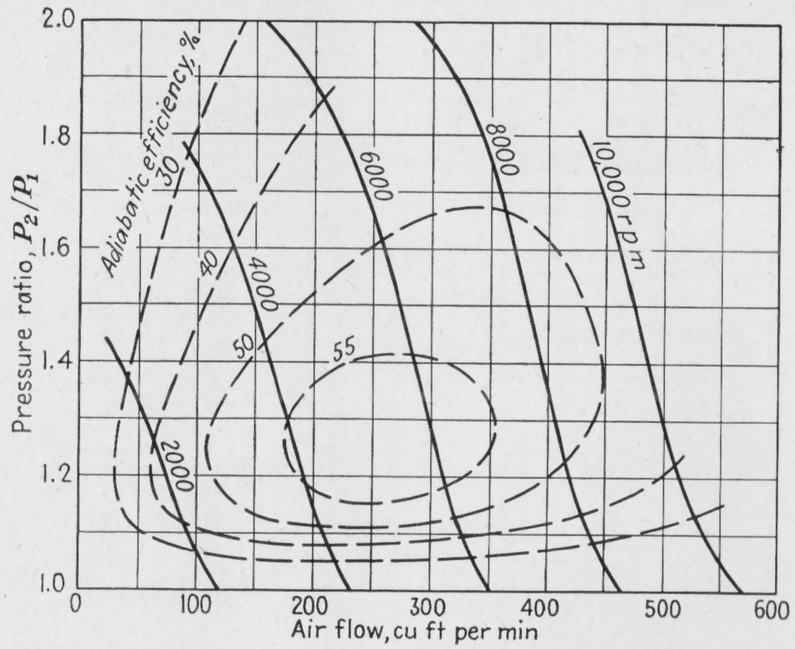

FIG. 189. Chart, or "map," showing the performance characteristics of a Roots blower for an automotive engine.

efficiency are superimposed in dotted form on the constant-speed curves to give a contour-map effect and thus clearly show the speeds and flow rates giving high efficiency. Although this type of chart may seem at first to be quite complex, some experience in working with it will show how inadequate the more simple curves of Fig. 188 really are.

Lysholm Compressors. The Lysholm-type compressor resembles a Roots blower in that air is trapped between two parallel contoured rotors. It differs from the Roots blower in that the inlet and outlet ports are not rectangular openings on either side of the housing but are instead specially shaped ports placed at opposite ends in such a way that the air is compressed between the rotors before the exhaust port is uncovered.[4] This feature is evident in Figs. 190 and 191, which show both an end view and longitudinal sections through a typical unit.

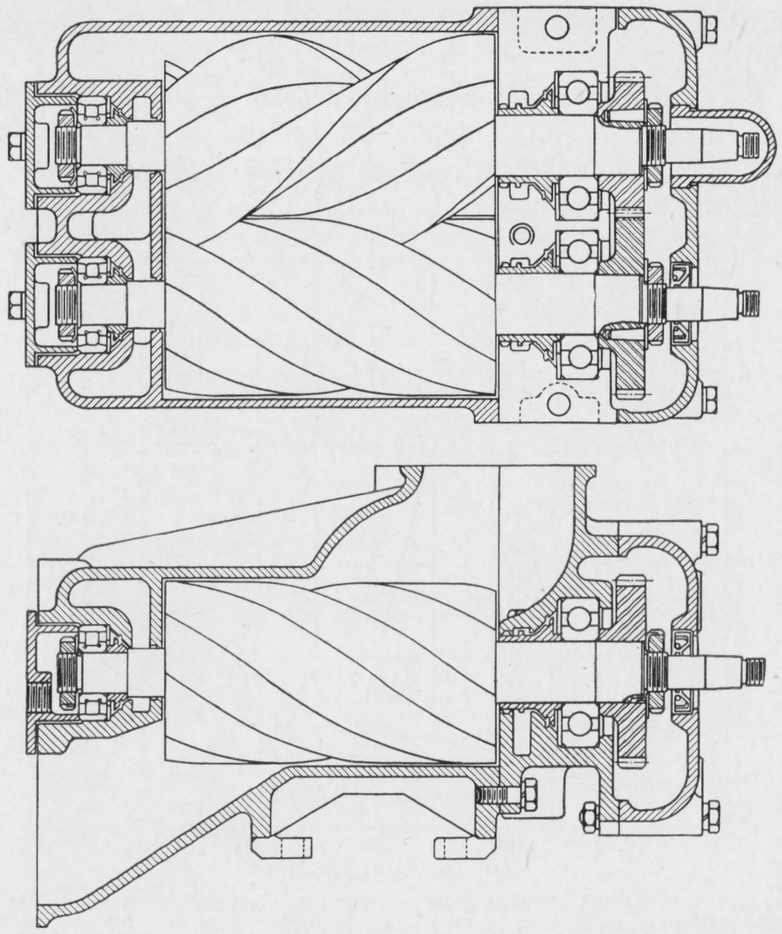

FIG. 190. Sections through a Lysholm compressor showing the rotors and the arrangement of the ports. (*Elliott Company.*)

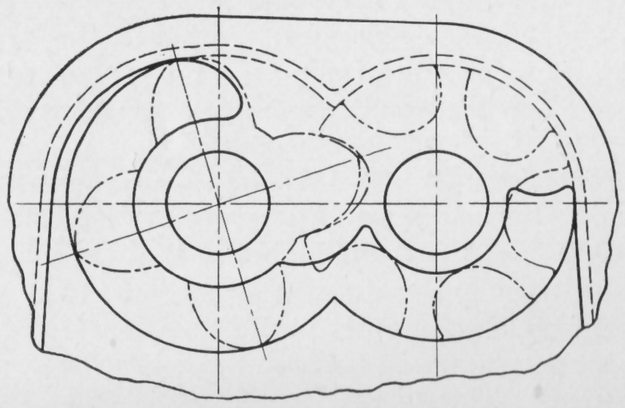

FIG. 191. Cross section through the rotors for the Lysholm compressor of Fig. 190 showing their shape as well as the shape of the intake port. (*Elliott Company.*)

The Lysholm compressor is essentially a constant-compression-ratio machine in that the exhaust-port shape determines the extent to which the air is compressed between the rotors before the port is uncovered. The compressor will operate quite well at other than the design compression ratio, however. If the delivery pressure is higher than the design value, the last bit of compression will be by backflow compression, as in a Roots blower. If the delivery pressure is below the design value, the air will reexpand when the exhaust port

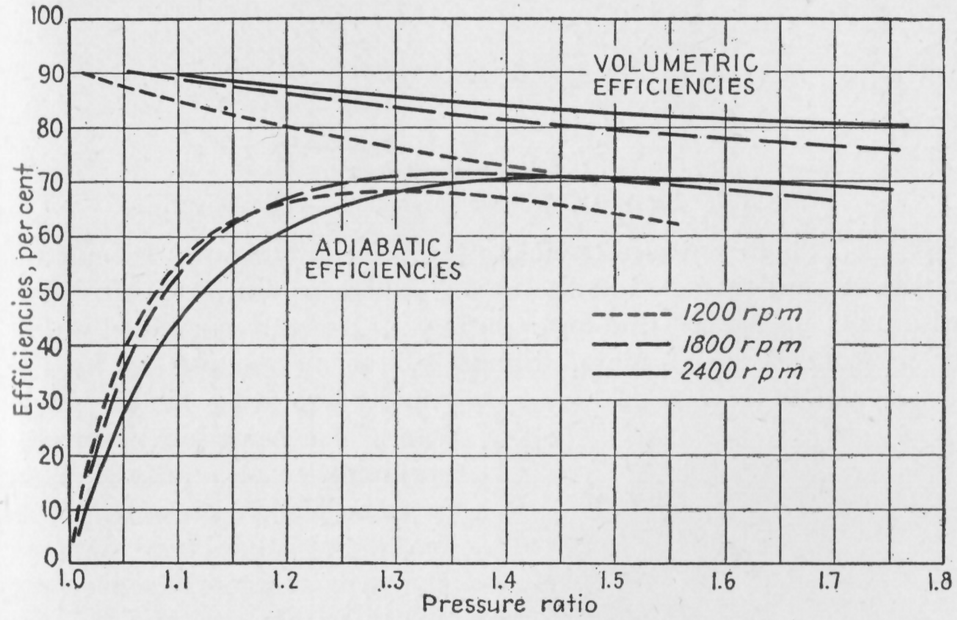

FIG. 192. Effect of pressure ratio on the adiabatic and volumetric efficiencies of a Lysholm-type compressor. (*Elliott Company.*)

is uncovered. A set of curves similar to those in Fig. 188 for a Roots blower are given in Fig. 192 for a Lysholm compressor. These demonstrate that the Lysholm compressor has a much wider practicable operating range than might at first be supposed. Note the similarity between its characteristics and those shown in Fig. 188 for a Roots blower.

Compression in Centrifugal Compressors. The mechanism of compression in a centrifugal compressor is more difficult to visualize than in a positive-displacement machine. Because of this and because of several important additional relations that apply, some expressions for the performance of centrifugal compressors are presented in this section.

The path of the air flowing through a centrifugal supercharger

may be traced in either Fig. 29 or Fig. 224. A rigorous derivation of the pressure rise ideally attainable involves some rather complex fluid mechanics. A rough analogy will serve for the purposes of this text, however. Consider first the pressure distribution over the surface of a streamlined body moving rapidly through the air. As indicated in

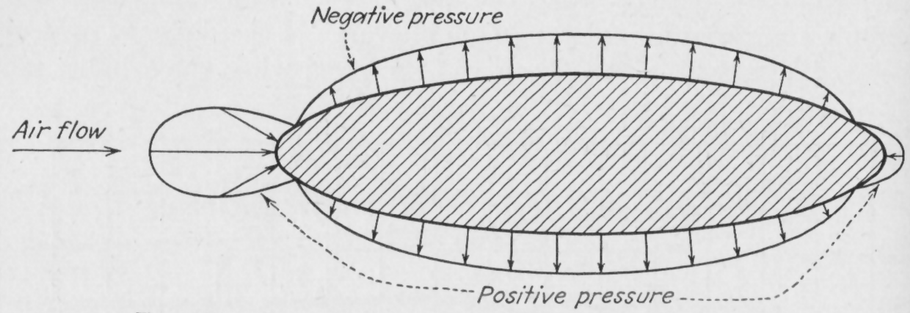

FIG. 193. Pressure distribution around an elliptical strut.

Fig. 193, a high pressure (equal to the velocity pressure) is built up in the stagnation region at the nose. A high pressure region is likewise built up against the lower surface of a highly cambered airfoil or in front of a flat plate. Similarly, the air scooped up by the rapidly rotating inlet guide vanes of a centrifugal-compressor impeller also undergoes an increase in static pressure. This may be visualized by examining the supercharger impeller inlet shown in Fig. 194 and the diagram in Fig. 195 showing the relative velocities at the impeller inlet. The impeller continues to "push" on the air and force it to accelerate tangentially as it flows radially outward between the vanes so that by the time the air reaches the periphery its tangential velocity is equal to the tip speed of the impeller. By acting on the air in this way, the impeller brings about a further rise in static pressure. For an ideal incompressible fluid flowing through a radial-vaned impeller in which the flow-passage areas at the inlet and outlet are equal, the static-pressure rise in the impeller is equal to $\rho V^2/2$, where ρ is the fluid density and V is the change in the tangential velocity of the air

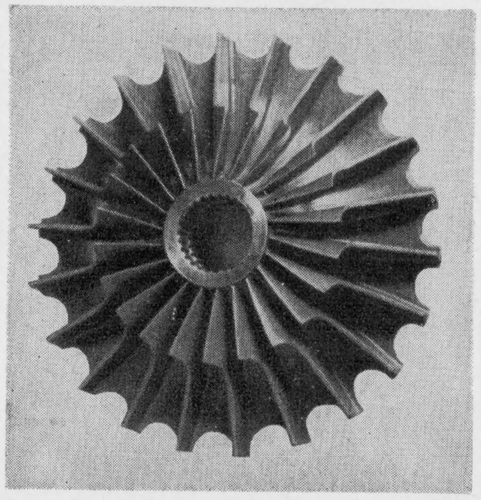

FIG. 194. Impeller for a centrifugal supercharger.

in passing through the impeller. Since the tangential velocity of the air at the impeller inlet is normally zero, for practical purposes V can be taken as equal to the tip speed of the impeller.[5] If the impeller vanes are not radial—as is sometimes the case—the reasoning becomes more complex but the same basic relations between changes in static pressure and changes in fluid velocity still hold.

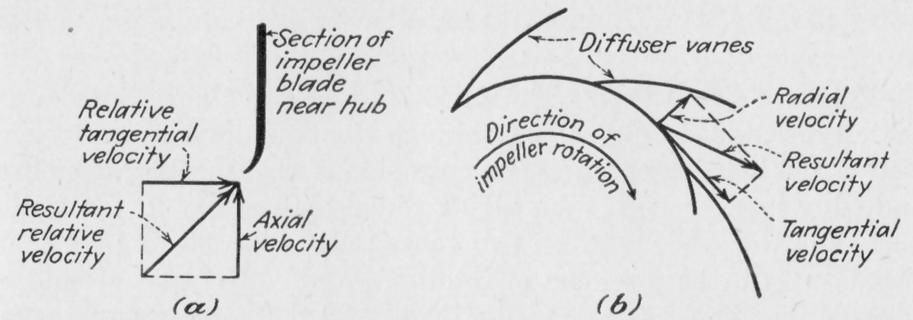

FIG. 195. Components of the relative velocity for the air stream passing through a centrifugal compressor for (*a*) an impeller inlet guide vane, and (*b*) the inlet to the diffuser vanes.

An additional static-pressure rise can be obtained after the fluid leaves the impeller by converting its high tangential-velocity energy into pressure energy. A set of expanding passages called a *diffuser* can be arranged to slow down the high-velocity stream with a minimum of eddy losses. The pressure rise ideally attainable is the same as that from the acceleration of the fluid in the impeller, viz., $\rho V^2/2$, where V may again be taken as equal to the tip speed of the impeller if the vanes are radial. If compressibility effects on the density are small, a first approximation to the total ideal pressure rise for the compressor is the sum of the two, or, algebraically,

$$\text{Ideal pressure rise} = \text{impeller } \Delta p + \text{diffuser } \Delta p$$
$$= \rho \frac{V^2}{2} + \rho \frac{V^2}{2} = \rho V^2$$

The ideal power input to the impeller may be determined from much the same considerations as those used above for the pressure rise. The only air forces acting on the impeller affecting the power input are the tangential forces on the blades. The acceleration of the air from a zero tangential velocity to the tip speed of the impeller involves a change of momentum. The force required to produce this change in momentum is provided by the impeller. Thus the reaction force acting on a radially vaned impeller is given by

$$\text{Reaction force} = (\text{mass})(\text{change in velocity}) = \frac{WV}{g}$$

where W = weight flow of air, lb/sec
V = impeller-tip speed, ft/sec

The power required to drive the impeller is the product of this force and the distance that it moves per unit of time. Since the latter is also equal to the impeller-tip speed, the power required becomes

$$\text{Hp input to impeller} = \frac{WV^2}{550g} \tag{27}$$

Pressure Coefficient. The principal losses in the impeller are of three types, viz., back leakage through the boundary layer along the wall of the housing opposite the open side of the impeller, circulation and eddy losses caused by a higher radial air velocity in the trailing portion of the space between two vanes than in the leading portion of that space (in the direction of rotation), and eddy losses around the edges of the vanes at the impeller inlet. All these losses act to reduce the actual pressure rise but do not change the power input from the value derived above. Thus another measure of the efficiency of a centrifugal compressor is the ratio of the isentropic power required for compression between a given set of pressure limits and the power required on the basis of Eq. (27). This measure of efficiency is called the *pressure coefficient* and is used only in connection with centrifugal compressors. It may be derived as follows:

$$\text{Press. coeff} = \frac{\frac{1}{550}(k/k-1)WRT_1Y}{WV^2/550g} = \frac{6{,}088T_1Y}{V^2} \tag{28}$$

Characteristics of Centrifugal Compressors. The characteristics of a centrifugal compressor are much different from those of the constant-displacement machines described earlier in this chapter. Figure 196 shows a set of curves that completely describe the performance of a centrifugal-type supercharger. Note that, instead of the pressure ratio changing little with speed as in the Roots or Lysholm machines, it varies approximately as the square of the speed as indicated above for the ideal centrifugal compressor. [As shown in Eq. (28), the Y factor varies directly as the square of the tip speed, and the pressure ratio accordingly.] In the range of high adiabatic efficiencies, Fig. 196 shows that the pressure ratio at constant speed remains substantially constant in the centrifugal compressor, while the flow through the unit varies. This is exactly the opposite of the corresponding characteristic of the constant-displacement types of compressor, for in them the flow rate remains substantially constant in the high-efficiency range while the pressure ratio varies. It follows from this

that one type of compressor is often better suited to one variety of application and the other type to another.

The ratio of the maximum to the minimum air flow rates at which good efficiency can be maintained is known as the *range* of the machine. Centrifugal compressors that are characterized by a wide operating range are likely to have a low efficiency although very careful design and construction may make both good range and good efficiency possible. The reason that the efficiency tends to peak rather sharply is that the relative velocities at both the impeller and the diffuser inlets

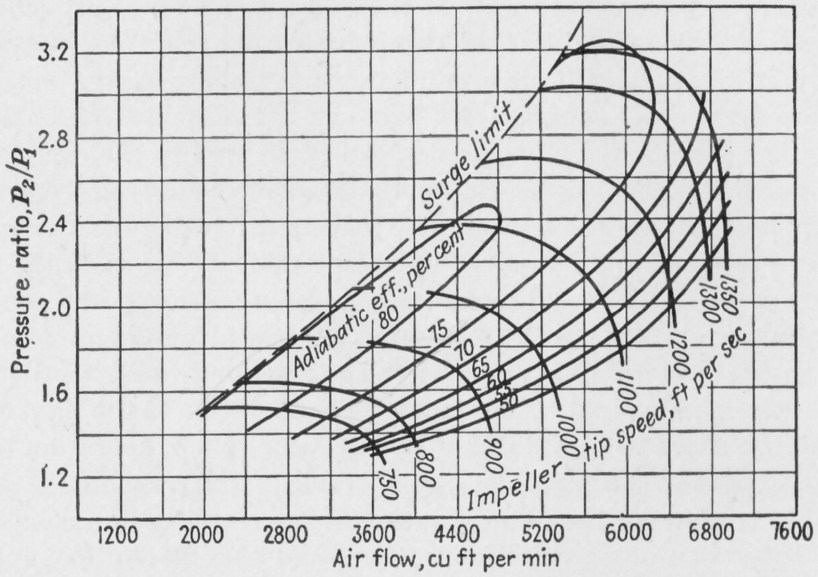

FIG. 196. Performance characteristics of a centrifugal supercharger. (*Campbell and Talbert, Trans. SAE, Vol.* 53, 1945.)

must be such that eddies are minimized if good efficiency is to be attained. This effect may be visualized by examining Fig. 195. At any given impeller speed, the tangential-velocity components are fixed. If the air flow is varied, both the axial component at the impeller inlet and the radial component at the impeller outlet are changed accordingly. At the design value, the air flow rate will be such that the relative velocity components are tangent to both the impeller inlet guide vanes (also called *buckets*) and the inlet portion of the diffuser vanes. At any other flow rate the air streams will strike the impeller and diffuser vanes at an angle. Eddy losses inevitably result.

One objectionable characteristic of centrifugal superchargers is their tendency to *surge*. Note that the constant-speed curves of Fig. 196 are discontinuous at a line labeled "Surge limit" at their left ends. The reason for this discontinuity is that a further reduction in flow

would throw operation into an unstable range in which violent pressure and flow fluctuations occur. These fluctuations are caused by the fact that any small reduction in flow in the surge range brings about a reduction in efficiency, which, in turn, brings about a reduction in pressure rise and hence a further reduction in flow, etc., until the flow drops to an abnormally small value. Such a condition cannot be maintained, however, for the rapidly spinning impeller sets up a pressure differential that induces a very large flow. The resistance of the system will not permit the large flow to continue for more than an instant, however, and therefore the flow rate again drops below the surge limit. The events in the complete surge cycle take place very rapidly, the frequency of the flow fluctuations usually being 1 to 60 cps.

Consideration of supercharger behavior during operation on a portion of the characteristic constant-speed curve well to the right of the surge point will also help to clarify the concept of supercharger surge. Any momentary increase in air flow caused by a disturbance in the air-flow system would cause the operating point to move to the right on the curve. This would mean a loss in both efficiency and pressure ratio and would reduce the pressure differential acting to force air through the system. Consequently, the air flow would tend to fall off to the original value. Thus compressor operation to the right of the peak of the curve will be stable. By the same token, operation to the left of the peak of the curve must be unstable. The reasoning involved is similar to that for the determination of the stability of a solid body under the action of gravity. It is sometimes possible, for example, to stand an egg on end on a flat surface; but any slight disturbance will tend to lower its center of gravity, and hence this position is unstable. If, on the other hand, an 0.5-in.-diameter hole is drilled in the flat surface and the egg stood on end in it, any small angular displacement will tend to raise rather than lower the center of gravity of the egg, and hence the condition will be stable. The basic elements of stability in a given supercharger are not greatly different from those of an egg in this respect, for it is usually impractical to try to operate in the unstable, or surge, range of the compressor.

Curves of the type shown in Fig. 196 often are given in somewhat different terms because of the large effect of air inlet temperature on centrifugal-compressor performance. Instead of the values for air flow and tip speed being used directly for a particular value of air inlet temperature, the effect of that factor can be eliminated by multiplying both the tip speed and the air flow rate by the square root of the absolute temperature to obtain a new pair of parameters. A chart

such as that in Fig. 196 may be used equally well, however, if proper corrections are made for air inlet temperature. Allowance for temperature effects may be made by multiplying both the air flow and the tip speed at any given point on the chart by the square root of the ratio of the absolute air inlet temperature for which the chart was constructed to the absolute air inlet temperature for the condition in question. That is, the pressure ratio, flow, and efficiency of the supercharger of Fig. 196 at an air-flow rate of 4,800 cfm, a tip speed of 1,000 ft/sec, and an inlet air temperature of 100°F could be determined by multiplying the tip speed by $\sqrt{520/560}$, or 0.96, to obtain 960 ft/sec. The flow indicated by the 1,000 ft/sec curve should also be multiplied by 0.96 to obtain 4,608 cfm. The pressure ratio and efficiency at 100°F would then be given by the chart at these temperature-corrected values for flow and tip speed.

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Problems

1. If the air consumption of a supercharged engine runs 20 lb/min and the pressure ratio of the supercharger is 1:2, determine the power input to the supercharger and the air temperature rise, assuming pressure and temperature coefficients of 0.70. Repeat for a pressure ratio of 2.0:1, and compare.

2. A four-stroke-cycle unsupercharged diesel engine is to be supercharged and a high valve-overlap camshaft installed in an effort to increase the engine output. If the supercharger is to have a pressure ratio of 1.2:1 and the engine compression ratio is 15:1, what is the percentage increase in imep ideally available from scavenging the clearance volume and increasing the cylinder pressure at the beginning of the compression stroke? Assume that the maximum allowable fuel-air ratio is to be kept constant. Repeat for a supercharger pressure ratio of 2.0:1.

3. Assuming that the engine of Prob. 2 is provided with the supercharger of Prob. 1, analyze the results of Probs. 1 and 2, and comment on the extent to which the increase in output justifies the increased power required by the supercharger. Assume that the volume of fresh scavenging air blown through the cylinder and out the exhaust port is equal to four times the clearance volume.

4. Determine the ideal work input per pound of air for a Roots blower operating with backflow compression but no back leakage through the rotor clearances. Compute values for pressure ratios of 1.2, 2.0, and 4.0:1, and compare with corresponding isentropic work inputs.

5. Compare the effectiveness in terms of percentage increase in air density for (*a*) supercharging an engine to a pressure ratio of 2.0:1 with an adiabatic efficiency of 0.77; (*b*) supercharging as in (*a*) and then, by cooling the air, reducing its temperature rise to only 30 per cent of that in (*a*).

6. Assuming that the volumetric efficiency (based on intake manifold conditions) of a diesel engine is practically independent of speed or the amount of supercharging, determine the increase in engine output obtainable with a Roots-type supercharger at 600 and 1,200 engine rpm if the supercharger gives a pressure ratio of 2.0:1 at 1,800 engine rpm. Repeat for a centrifugal supercharger that also gives a pressure ratio of 2.0:1 at an engine speed of 1,800 rpm. Take the supercharger as being directly geared to the engine in both cases. Note that the pressure rise for the centrifugal compressor will vary widely with rpm, while that for the Roots blower will not.

CHAPTER XIII

PERFORMANCE ANALYSIS

Performance analysis is one of the most fascinating fields of engine work. A nice balance of theoretical background, familiarity with engine characteristics, and engineering judgment are necessary to interpret properly any type of test data, to prepare performance estimates for an engine in the design stage, or to prepare performance specifications from test data. Most of the material covered earlier in this text will have a bearing on the subject either directly or indirectly, but that in Chaps. IV to VI, Combustion, Cycle Analysis, and Fuel Metering and Injection, and Chap. XII, Supercharging, will prove to be especially fundamental to an understanding of the problems involved. By proper use of these fundamentals it is possible to relate engine speed, displacement, volumetric efficiency, air flow, fuel flow, ihp, fhp, bhp, and bsfc. This sort of analysis is essential to the evaluation of such items as the possible effects of a proposed change in design before it is tested or the actual effect of such a change indicated by comparative test data taken before and after the modification.

This chapter presents in a general fashion the basic relationships employed in performance-analysis work, with particular reference to the more widely used engine types. The nature of and the reasons for exceptions to the somewhat idealized characteristics presented are also mentioned but are not treated at length partly because the deviations are usually small and partly because the problems were considered too specialized to be of general interest.

ENGINE AIR CONSUMPTION

Air Consumption and IHP. By far the most important factor determining engine power capacity is engine air consumption. In spark-ignition engines in which the fuel-air ratio is normally constant, the ihp is directly proportional to the air consumption, usually expressed in pounds per hour. In diesel engines the peak permissible power output at any given speed is also directly proportional to the engine air flow, although, of course, the output at part load is not directly dependent on the air flow rate, for control is obtained by varying the fuel-air ratio. Perhaps the best way to relate air consump-

tion to ihp is to base it on the rate at which heat is released from the fuel as it is fed to the engine. On this basis, the ihp is given by

$$\text{ihp} = \frac{(\text{thermal eff})(\text{air cons, lb/hr})(F/A)(\text{heat from fuel, Btu/lb})}{2{,}545} \quad (29)$$

It was shown in Chap. V that the thermal efficiency at a constant fuel-air ratio will be a function primarily of compression ratio if the spark (or injection) advance is properly adjusted. Therefore, in a given engine with a fixed compression ratio and operating at a fixed fuel-air ratio, the ihp will be directly proportional to the air flow for any given set of conditions. From this it follows that the *indicated specific air consumption* (isac), in pounds of air per ihp, should be a constant for any given fuel-air ratio and compression ratio. The specific air consumption is widely used in spark-ignition-engine performance-analysis work because it is such a convenient and reliable parameter.

Effect of Fuel-Air Ratio on ISAC. It was shown in Chapts. IV and V, Combustion and Cycle Analysis, that certain characteristic variations in thermal efficiency and specific fuel consumption are to be expected as the fuel-air ratio is varied. The nature of these effects on the isac, isfc, and ihp for a representative spark-ignition engine are shown in Fig. 197. The shape of curves such as those in Fig. 197 will vary somewhat with rpm, mixture distribution, volumetric efficiency, spark timing, and valve overlap; but both the values given and the shape of the curves shown are typical for well-designed spark-ignition engines in good adjustment.

Characteristic curves for isac and isfc for a diesel are not given here, for isac is not often used in diesel performance analysis work. This is because the most convenient parameters are those which stay constant. For spark-ignition work where the fuel-air ratio remains substantially constant, the isac is constant and hence is a useful parameter. On the other hand, in diesel work, the isac varies widely, while the isfc changes only to a small extent, and that as an almost straight-line function of fuel-air ratio. This effect is indicated by Fig. 81 (page 144), which shows data for the indicated thermal efficiency of a diesel engine as a function of fuel-air ratio. Note that in the normal operating range of fuel-air ratio for diesel work the isfc would run from about 0.28 to around 0.35 lb/ihp-hr.

Effect of Compression Ratio on ISAC. As mentioned previously, the improvement in thermal efficiency with increasing compression ratio effects a reduction in the isac. The nature of this effect is shown in Fig. 198 for the best-power fuel-air ratio and spark advance.

The curves in Fig. 197 and 198 may be used to estimate the isac of any spark-ignition engine. The magnitude of the error involved in such an estimation is a function of the difference in cycle-efficiency ratio between the engine in question and that used as a basis for the curves. In addition, in supercharged engines, any possible by-pass

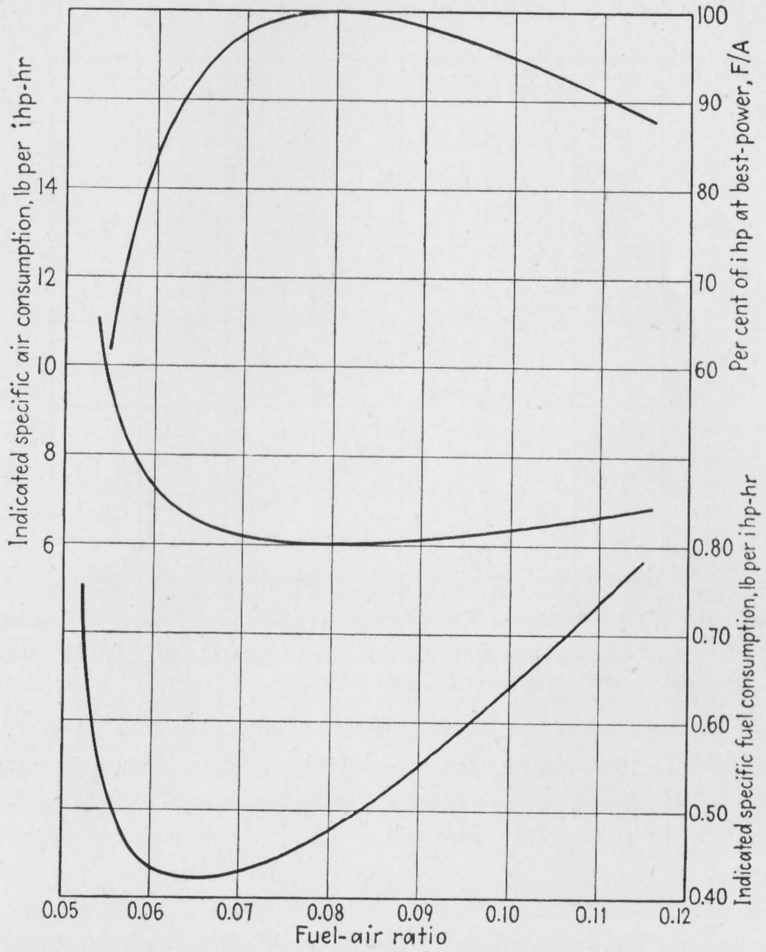

FIG. 197. Effect of fuel-air ratio on the isac, the isfc, and the ihp of a six-cylinder L-head passenger-car engine with a 6.7 : 1 compression ratio. (*Case Institute of Technology.*)

flow of fresh charge during the valve-overlap period will introduce a further error. While both these sources of error are susceptible to the sort of detail analysis indicated in Chap. V, it is usually more convenient to prepare characteristic curves of isac or isfc from test data for the engine under consideration. These curves may then be used for engine performance work without troubling about the individual

effects of cycle efficiency, etc., except to make certain from test data that the isac and isfc curves do not change too much with rpm.

VOLUMETRIC EFFICIENCY

Relationship between Engine Air Consumption and Volumetric Efficiency. The air consumption of an engine is most easily handled in analytical work by considering the engine as a constant-displacement air pump. As demonstrated in Chap. II, the amount of air induced by a four-cycle engine is then given by

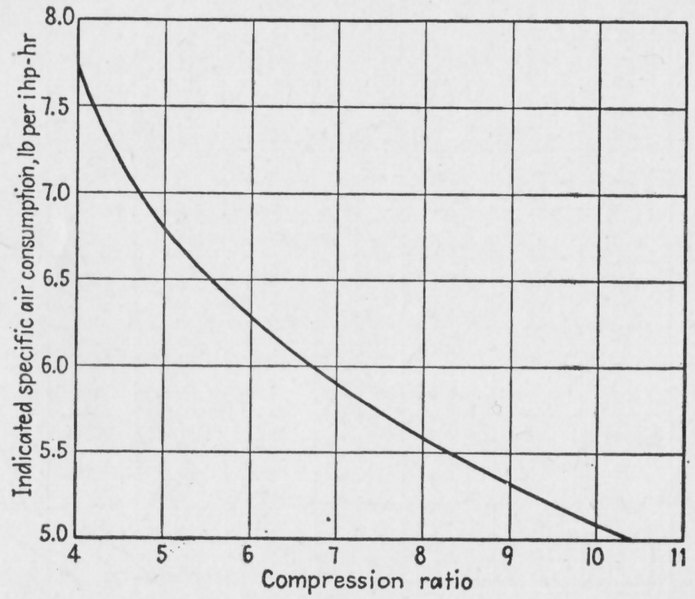

FIG. 198. Effect of compression ratio on the isac of typical spark-ignition automotive engines for operation with a fuel-air ratio of 0.080.

$$\text{Air flow, lb/hr} = \frac{(\text{volumetric eff})(\text{piston disp., in.}^3)(\text{rpm})(\text{air density, lb/ft}^3)(60)}{2 \times 1{,}728}$$

The air density is almost always taken as atmospheric in specification work, although the intake manifold density may be used for certain specialized analyses *e.g.* as for supercharged engines or for part-throttle conditions in any type of spark-ignition engine. This is reasonable; for only full-throttle operation is important in most specification work, and under those conditions the effects of the pressure losses through the air cleaner and/or carburetor are just as important as those across the valves, and thus they may as well be lumped together. If the volumetric efficiency is based on atmospheric conditions, it is determined only from full-throttle data. In nonspecification analysis work, it may be desirable to separate the effects of the many operating variables such as the pressure drop across the carburetor and the pressure rise through a supercharger and consider only the air-pumping performance of the cylinder between the intake and the exhaust ports. In that case the volumetric efficiency may be determined from either full-throttle or part-throttle data on the basis of the intake manifold density.

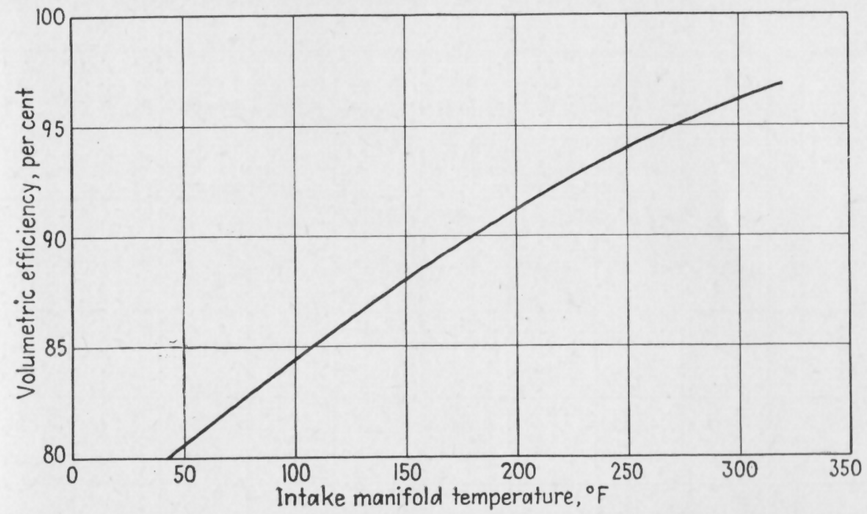

FIG. 199. Effect of intake manifold temperature on the volumetric efficiency of an aircraft engine with ample valve port area and an average interior cylinder-head metal temperature of 350°F.

The characteristic called *volumetric efficiency* depends on many factors. The interrelationships of these quantities are so many and varied that no one simple relationship can be given to include them all. Each of the more important factors will be discussed, its ideal effects considered, and comparisons made with curves obtained from test data.

Effects of Heat Transfer from Combustion-chamber Walls. As pointed out in Chap. V, heat transfer from the combustion-chamber walls to the fresh charge causes a marked increase in charge temperature and hence a reduction in charge density and volumetric efficiency.[2] This effect does not vary greatly with speed, but it is markedly affected by intake manifold temperature. The effect is most evident in a case in which the volumetric efficiency is computed on the basis of manifold air density. It will then be found that the volumetric efficiency actually increases with intake manifold temperature. This happens because increasing the intake manifold temperature serves to reduce the temperature differential between the fresh charge

and the combustion-chamber walls. This, in turn, decreases the heat transfer from the walls to the fresh charge and hence produces less of a reduction in density during the intake stroke. Figure 199 shows the effect of manifold temperature on the volumetric efficiency of an engine in which the average combustion-chamber-wall temperature ran about 350°F, including the effects of high exhaust-valve temperatures. The loss in volumetric efficiency caused by heat transfer from the combustion-chamber walls may also be deduced from Fig. 199 if

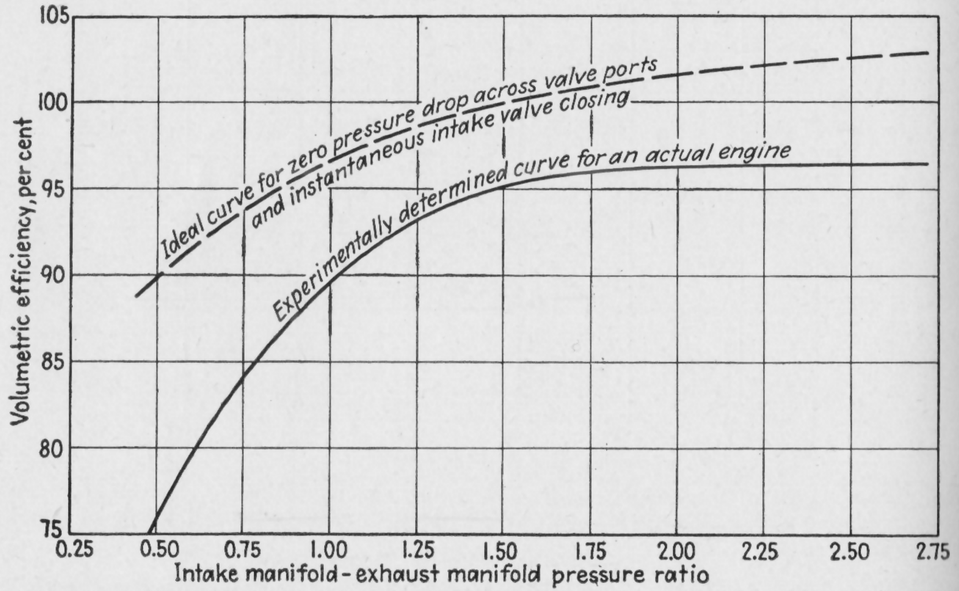

FIG. 200. Effect of the ratio of intake manifold pressure to exhaust back pressure on the volumetric efficiency of the supercharged aircraft engine of Fig. 199.

the loss for a 350°F mixture temperature were taken as zero. (There should be no heat transfer from the combustion-chamber walls to the fresh charge when their temperatures are the same.) From Fig. 199 it is evident that heat transfer from the combustion-chamber walls causes a loss of about 13 per cent at what might be called "normal" manifold temperatures, viz., 90 to 120°F.

Effects of Intake-manifold–Exhaust Back-pressure Ratio. The heating effect of residual exhaust gas on the fresh charge was considered in Chap. III, and the attendant loss in volumetric efficiency was computed for the ideal case under full-throttle conditions. While the loss is only 5 or 10 per cent if the intake and exhaust manifold pressures are approximately equal, it becomes larger very rapidly as the ratio of intake to exhaust pressure is reduced, as, for example, under throttling conditions. As was shown in Chap. III, this loss can be computed readily for the idealized cycle using the actual working

fluid. Figure 200 shows a curve for the volumetric efficiency as a function of the intake-manifold–exhaust back-pressure ratio for both an actual engine and the idealized case as treated in Chap. III. It should be mentioned that curves of this sort for actual engines are inclined to vary widely in slope with variations in valve timing, particularly with changes in valve overlap. The curve shown is for an engine with virtually no valve overlap.

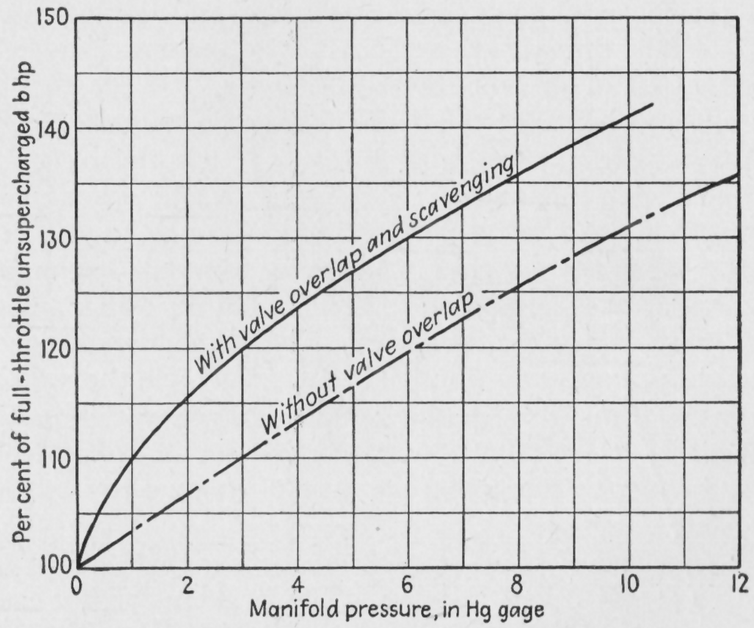

FIG. 201. Effect of a supercharge pressure on the output of a diesel engine having a large amount of valve overlap.

In this connection, it would be well to point out the nature of the possible benefits of supercharging in diesel engines, where a considerable amount of valve overlap can be used. By maintaining an intake manifold pressure higher than the exhaust manifold pressure, the residual exhaust gas in the clearance volume can be blown out and replaced with fresh charge. Blowing air through the clearance volume of a diesel during the valve-overlap period also serves to cool the combustion-chamber-wall surfaces. Both factors yield an increase in the obtainable engine output by increasing the quantity of fresh air available to burn the fuel each cycle. Since volumetric efficiency as previously defined cannot be measured readily when air is blown through the engine for scavenging purposes—the quantity of fresh charge flowing out through the exhaust port during the valve-overlap period being unknown—an indication of the increase in charge density resulting from supercharging may be obtained by plotting power out-

put against scavenging pressure, *i.e.*, supercharging pressure. Figure 201 showing a curve of this type demonstrates the marked increase in output obtainable from a small amount of supercharging and the proportionally lower returns from greater amounts. These benefits cannot be obtained in carbureted engines without an excessive loss of fuel in the fresh charge blown into the exhaust manifold with the scavenged residual gases. For this reason, lower valve overlaps are used in supercharged spark-ignition engines, and the increase in output is practically directly proportional to the amount of supercharging.

Effect of RPM on Volumetric Efficiency. Of the prime engine variables, speed has the most varied effects on volumetric efficiency. It has three different substantial effects. In the first place, the pressure drop through the induction passages—*i.e.*, the air cleaner, carburetor, manifold, and valve port—increases as the square of the air flow rate. Thus the pressure in the cylinder at the beginning of the compression stroke should be expected to fall off by an amount proportional to the square of the speed. The volumetric efficiency based on atmospheric conditions would, of course, fall off in the same manner. The pressure drop through the exhaust system varies in the same fashion as that in the induction system and acts to reduce volumetric efficiency further by increasing the ratio of residual to fresh charge in the cylinder.

A second effect of speed arises from the fact that engines are operated with a constant valve timing. The intake valve, for example, is usually set to close 60 deg or more after bottom center in high-speed engines. At low speeds not only is there plenty of time for the cylinder to fill with fresh charge, but there is also time for some of the charge that enters the cylinder before bottom center to be pushed back out into the intake manifold before the intake valve is completely closed. This is not the case at the higher speeds. If an appreciable amount of valve overlap is employed, it also has an effect that varies with speed. The valve timing adopted for any given engine is dependent on the intended engine application. Engines for such vehicles as heavy trucks and tanks have camshafts designed to give little valve overlap, a late exhaust-valve opening, and early intake-valve closing so that the engine will have a high volumetric efficiency and hence a high torque capacity at low speeds. This entails a considerable sacrifice in performance at high speeds, but it gives good starting and low speed lugging characteristics to the engine. Passenger-car engines have camshafts designed to give a peak volumetric efficiency at medium speeds with some sacrifice at both high and low speeds.

Racing-car and aircraft engines are designed to give their peak volumetric efficiency at high speeds, where it is most needed in that type of application.

A third possible effect of speed was mentioned in the chapter on manifolding—under just the right conditions, resonance of the air column in the intake manifold and cylinder may be excited by the intermittent flow conditions. If this occurs, a pronounced gain or

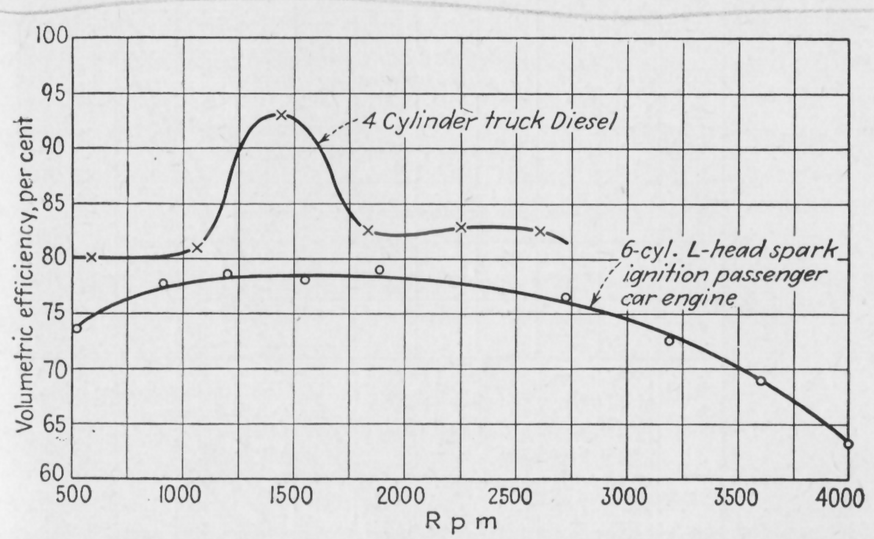

Fig. 202. Effect of rpm on the volumetric efficiency of two widely used models of automotive engine.

loss in volumetric efficiency will result, depending on whether the cylinder is in a region of high or low pressure at the time when the intake valve closes.

Curves of volumetric efficiency as a function of rpm are shown in Fig. 202 for two types of engine. Note that the curves fall off at high speeds in both cases. This is because of restrictions in the induction and exhaust systems. Note, too, the "hump" in the curve for the truck diesel engine—this is a typical indication of a resonance effect.

FRICTION HORSEPOWER

Components of FHP. Frictional losses in an engine are of two main types—mechanical friction and air-pumping losses. The mechanical losses include both boundary friction and fluid-film friction. The air-pumping losses arise from the pressure differential across the piston during both the intake and the exhaust strokes. That is, the cylinder pressure is usually below atmospheric during the intake stroke and above it during the exhaust stroke. In both cases

the crankshaft must do work on the piston during that phase of the process, for the crankcase pressure would be essentially atmospheric.

The mechanical frictional losses from both boundary and fluid-film friction increase with load, both being roughly proportional to the load. The larger coefficient of friction for boundary lubrication

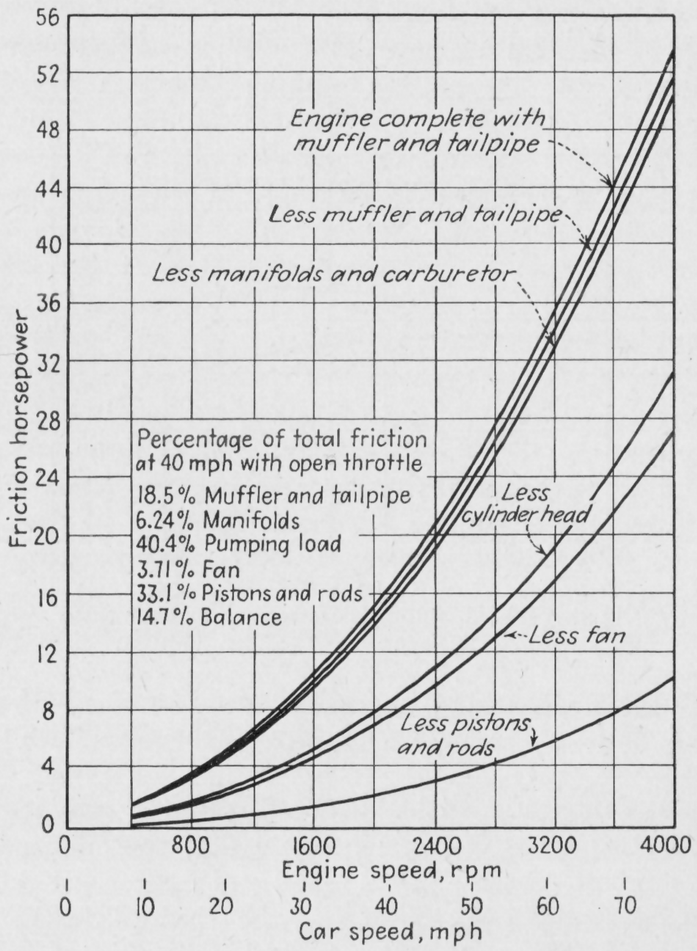

FIG. 203. Effect of rpm on the principal friction horsepower components for a six-cylinder passenger-car engine. (*Youngren, Trans. SAE, Vol.* 49, 1941.)

is an important factor in making piston and ring friction run from 30 to 50 per cent of the total mechanical friction, *i.e.*, much more than any other single set of parts. The next most important source of frictional loss is in the main- and connecting-rod bearings. The camshaft and valve gear entail certain small losses. The accessories and their drive shafts and gears may require a considerable amount of power, particularly the generator and fan. The distribution of the frictional losses to these various engine components is shown in Fig.

203. While the loss entailed by any one set of parts is usually not large, the cumulative effect is substantial, causing a power loss of 10 to 40 per cent of the full-throttle indicated output.[3,4]

Methods of Measuring FHP. While the fhp might be determined by subtracting the engine brake output from the indicated output determined from indicator cards, such a procedure not only requires complex and expensive instrumentation but is very tedious and time consuming. Further, the accuracy of the determination is poor at best, for the method depends upon finding a relatively small difference between two relatively large quantities, ihp and bhp. In practice, it has been found more convenient and more accurate to measure the power required to motor the engine with an electric dynamometer. Such a procedure involves a number of inherent errors, the more important of which are as follows:

1. The exhaust pumping loss is greater for motoring than for firing operation, for all the air charge must be pushed out of the cylinder by the piston under motoring conditions, whereas the greater part of the exhaust gas literally blows itself out of the cylinder during the exhaust-blowdown process under firing conditions.

2. The intake pumping loss is greater under motoring conditions because of the effect of heat transfer from the combustion-chamber walls. The amount of heat transfer is less for motoring than for firing operation, and hence the mass air flow rate and the pressure drop across the intake port are increased.

3. The temperature of the oil, particularly in the piston-ring belt, is lower for motoring operation, and thus the oil viscosity and hence the frictional losses are higher.

4. The gas loads on the pistons, rings, and bearings are much lower during the power stroke, and hence the mechanical frictional losses are reduced considerably for motoring operation.

Since some of the factors operate in one direction and some in the other, the net effect is such that they practically balance out in most engines. Since the fhp is usually determined only as a means of obtaining the ihp, even an error as large as 10 per cent in the determination of the fhp of an engine would generally not be serious. This follows because, if the fhp were, say, 20 per cent of the ihp, a 10 per cent error in the fhp would mean only a 2 per cent error in the ihp. Since the fhp can be determined by motoring with a much greater accuracy than ± 10 per cent, it is evident that motoring tests are as good for the determination of fhp as any much more elaborate method.

Some engineers feel that the error in the determination of fhp by

motoring can be reduced by operating the engine under firing conditions, briefly cutting the ignition, motoring for a few seconds, reading the dynamometer beam load, and then turning on the ignition again. This is not hard to do with certain types of dynamometer equipment; but as may be deduced from the preceding paragraphs, it is doubtful whether or not there is a gain in accuracy. In yet another technique, the loss in bhp when one cylinder is kept from firing is determined and the ihp calculated from that data. While no more accurate than the other methods, this last technique has the advantage that it can be carried out with any type of dynamometer, including water brakes. Although each technique has its own advantages and disadvantages, most test specifications call for a simple motoring test at full throttle after the engine has been thoroughly warmed up.

Effects of Speed and Load on FHP. Friction-horsepower curves obtained from motoring tests may be plotted against rpm on logarithmic paper to give essentially straight lines. This procedure not only improves the accuracy of interpolation and extrapolation but also makes it much easier to draw the curve. Curves of this sort are shown in Fig. 204 for two representative engines. Note that the slope of the curves in both instances is somewhat less than 2:1. Lines having slopes of both 1:1 and 2:1 are also sketched in for reference. The slope of the fhp curve can be used as a check on the experimental data—it normally runs from 1.45:1 to 1.85:1. That is, the fhp can be related to the rpm by the equation

$$\text{fhp} = C(\text{rpm})^k \tag{30}$$

where the value of K usually runs from 1.45:1 to 1.85:1.

Effects of Oil and Coolant Temperature on FHP. While both the oil and the coolant temperature have marked effects on the fhp at the lower temperatures, in the normal range of operating temperature (160 to 210°F) the effect of variations in oil and coolant temperature is practically negligible.

BRAKE HORSEPOWER AND SPECIFIC FUEL CONSUMPTION

The effect on indicated output of the more important operating variables has been discussed at various points earlier in this text. The net effects of all these factors on engine bhp and bsfc can now be considered by first finding the resultant indicated output and then deducting the frictional losses.[5]

Effects of Speed at Full Throttle. The technique of handling a great many performance-analysis problems can be nicely demonstrated by considering the effects of speed on the full-throttle output of an unsupercharged spark-ignition engine. The indicated output would be directly proportional to speed if the volumetric efficiency were constant. As shown in Fig. 205, allowance for volumetric efficiency yields a curve that peaks in the upper speed range of the engine.

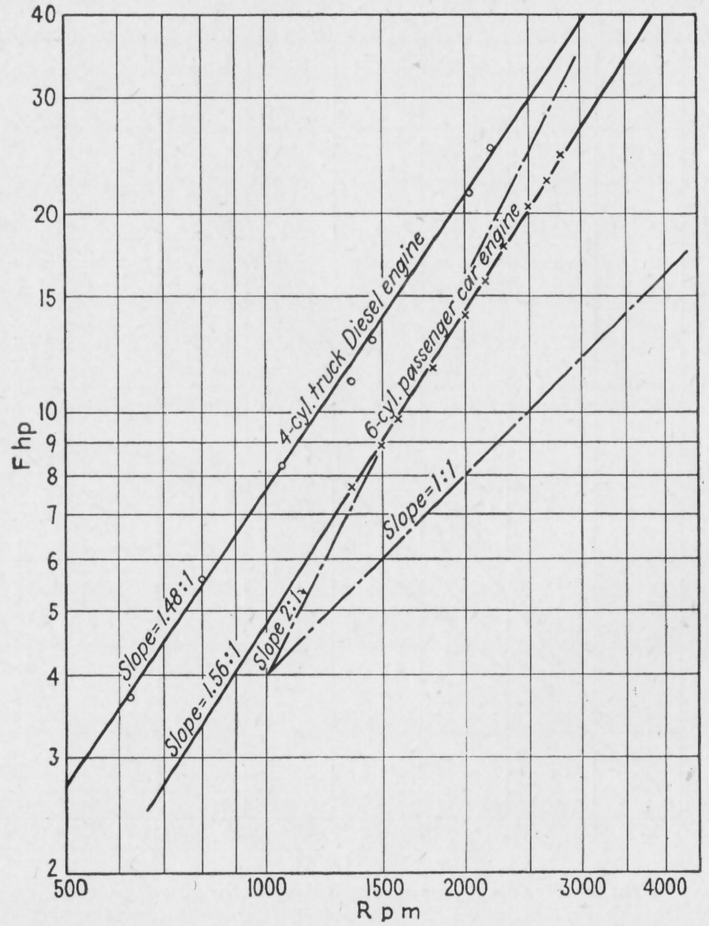

FIG. 204. Effect of rpm on the motoring friction horsepower of two representative engines having a piston displacement of 200 in.[3] Note the reference lines showing 1:1 and 2:1 slopes. (*Case Institute of Technology.*)

When frictional losses are deducted from the indicated output to give bhp, the resulting curve is lower and peaks at a lower speed than that for the ihp. This follows because the fhp increases more rapidly than in direct proportion to the speed. Since the isfc is practically unaffected by the volumetric efficiency, it is constant over the middle and upper portions of the speed range except insofar as the carburetor setting may give variations in the fuel-air ratio. A best-power mix-

ture was assumed in Fig. 205 to eliminate these effects. Because of the nonlinear characteristic of the fhp, the bsfc does not remain constant but tends to increase with speed. At the low-speed end of the range, losses caused by heat transfer to the combustion-chamber walls, early exhaust-valve opening (because the camshaft was designed for medium or higher speeds), and other factors bring about an increase

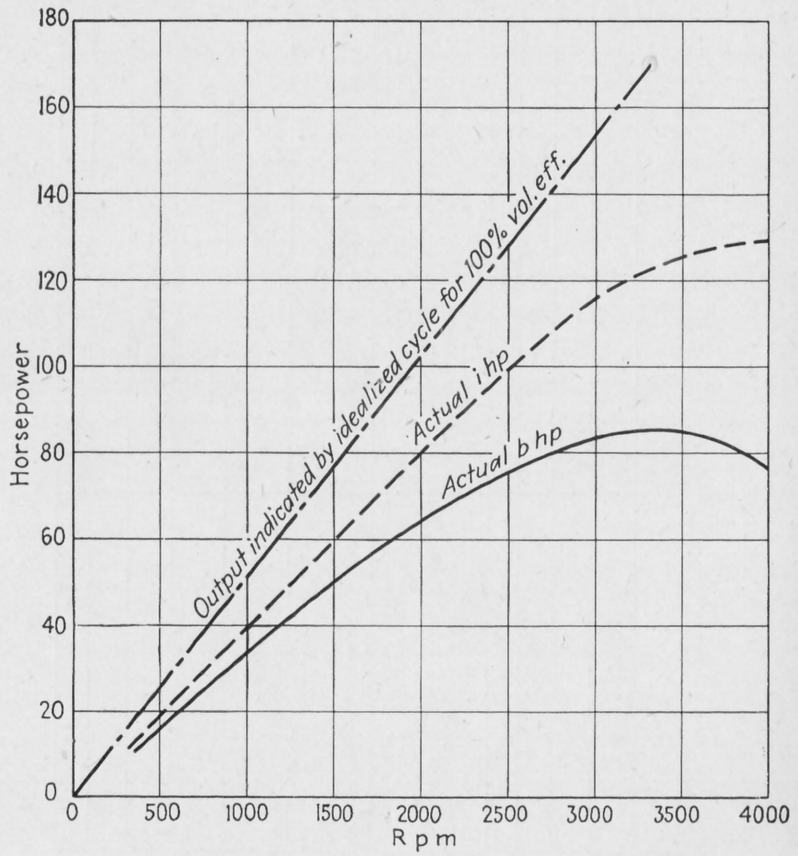

FIG. 205. Effect of rpm on full-throttle power output.

in isfc. This increase more than offsets the reduction in the power absorbed by friction at low speeds and brings about an increase in full-throttle bsfc at speeds below some optimum value.

Effects of Load on BSFC. It was pointed out early in this text that engine output at constant speed is controlled in diesel engines by varying the amount of fuel injected per cycle while leaving the air flow constant, whereas in spark-ignition engines the fuel-air ratio is maintained constant and the engine is throttled. It was shown that changing the load in this way had virtually no effect on the isfc for the idealized spark-ignition cycle but that the reduction in mean specific heat of the working fluid with an increase in load in diesel engines gives some increase in the isfc of the idealized cycle for a diesel operating on a constant-volume cycle. These effects were shown in Figs. 81 and 82 of Chap. V.

In relating the isfc to the bsfc for an actual diesel engine it has been found that, since the fhp changes relatively little with load at constant speed, the bsfc increases rapidly if the load is reduced much below the normal full-load value. These effects are shown in Fig. 206.

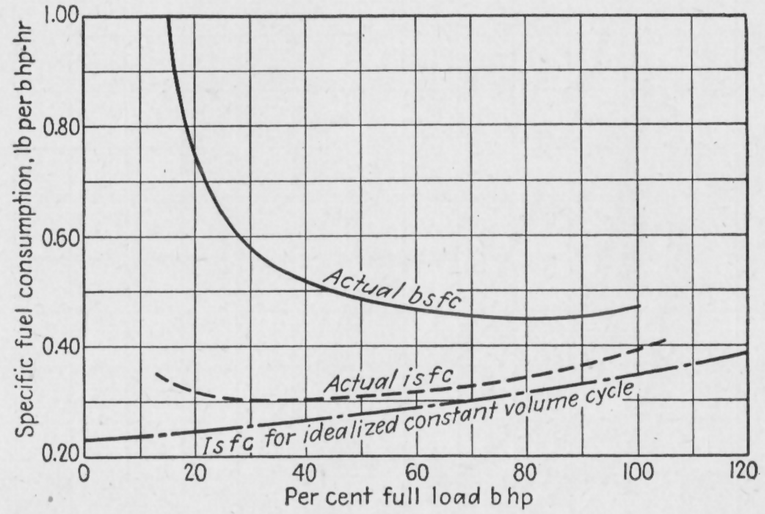

FIG. 206. Effect of load on the specific fuel consumption at constant rpm for a 200-in.^{3} displacement truck diesel engine.

The effect of load on the specific fuel consumption of a spark-ignition engine is much the same as in a diesel except for the small difference in the form of the isfc curve previously noted. The higher compression ratio of the diesel, of course, acts to give a somewhat lower bsfc than that given by a corresponding spark-ignition engine. The difference is not so great as might be expected from the isfc curves for the two engine types because the frictional losses are greater in the diesel. The higher cylinder pressures throughout the compression and power strokes increase piston and ring frictional losses in particular, while the higher ratio of peak pressure to imep means that larger and heavier parts are required. The latter, in turn, mean higher inertia loads and larger bearing areas—both acting to increase the frictional losses.

Effects of Speed on BSFC. Since most engines operate at part load and at any of a wide range of speeds, it is very important to present the effects of both speed and load on bsfc. This is often done by

plotting bsfc for each of a series of speeds as shown in Fig. 207. A set of curves of this sort is invaluable if a choice of possible transmission or rear-axle-gear ratios are being considered for a vehicle, for example. A corresponding set of curves for a spark-ignition engine would have much the same appearance as those for a diesel shown in Fig. 207.

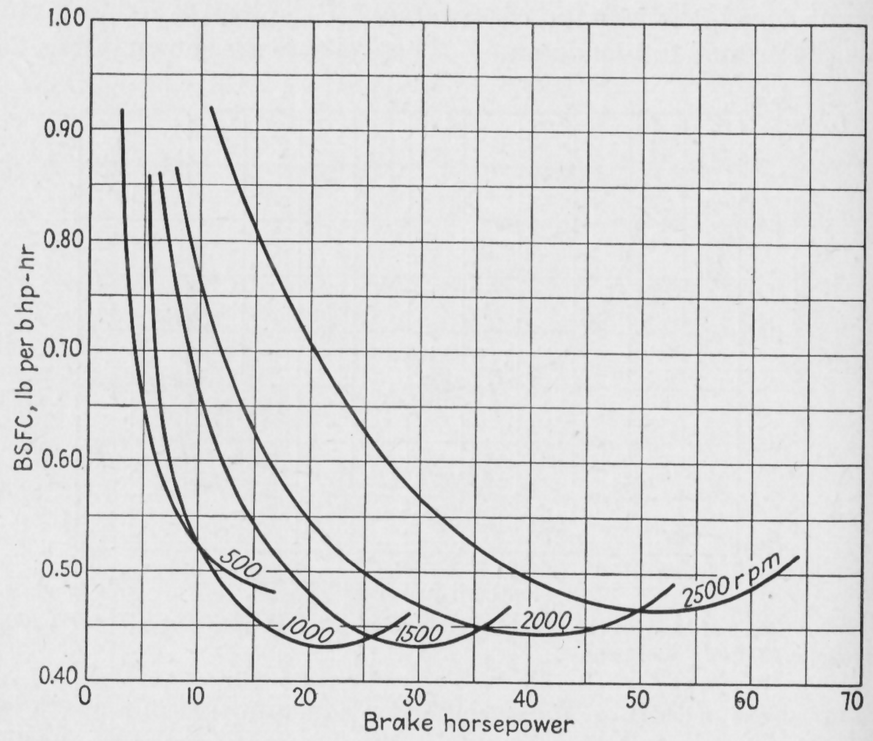

FIG. 207. Effect of speed and load on the bsfc of the same diesel engine as that of Fig. 206 (compression ratio = 14.5:1). (*Case Institute of Technology.*)

A more simple but less complete method of presenting engine performance is to give curves of bhp and bsfc for full-throttle conditions. Similar curves for a representative set of road-load conditions may also be included for automotive engines. A group of curves of this sort is shown in Fig. 208. These curves show how desirable it would be from the fuel-economy standpoint to have a variable-speed transmission that would make it possible to control a vehicle engine in the lower portion of the power range by keeping it near full-throttle irrespective of road speed or load while varying the engine speed to govern the output.

Effect of Supercharging. The power capacity of any given engine can be increased by supercharging, *i.e.*, by increasing the quantity of charge delivered to the cylinders each cycle. Engines often

require modification to take full advantage of supercharging. A diesel, for example, might require some change in the characteristics of its injection nozzles to obtain the proper amount of spray penetration. A spark-ignition engine may require some spark retard, richer mixtures, or a better fuel to avoid trouble with detonation. In the ideal

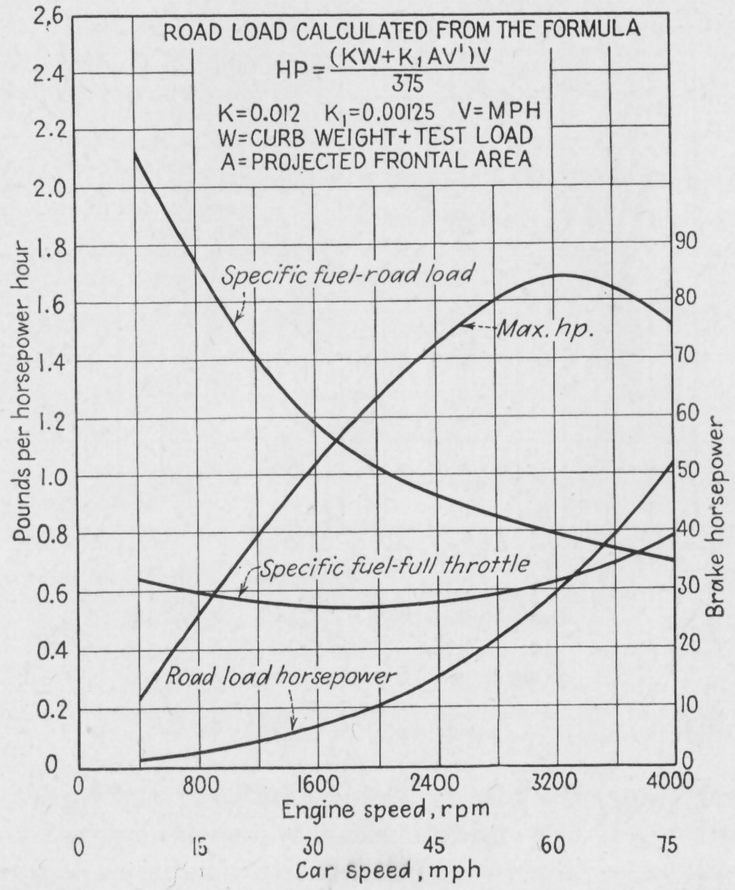

FIG. 208. Effect of speed on full-throttle and road-load bhp and bsfc. Curves plotted from dynamometer test data. The road-load bhp was computed using the formula at the top of the curve. (*Youngren, Trans. SAE, Vol.* 49, 1941.)

case, however, the indicated output should increase in direct proportion to the manifold air density. This ideal rate of increase can be realized by proper engine design. An even greater rate of increase can be obtained for small amounts of supercharging if, as was pointed out in the section on Volumetric Efficiency, valve overlap permits scavenging the residual gases from the cylinder. The greater part of the latter benefits can be realized with only a small amount of supercharging, as is evident in Fig. 201.

The power required to drive a supercharger was analyzed in the preceding chapter. Its magnitude is such that, if the supercharger is driven from the engine crankshaft, the ratio of net bhp to ihp tends to decrease somewhat with the amount of supercharging even if the supercharger-engine combination is very carefully designed. Thus, although the power output of a given engine can be greatly increased

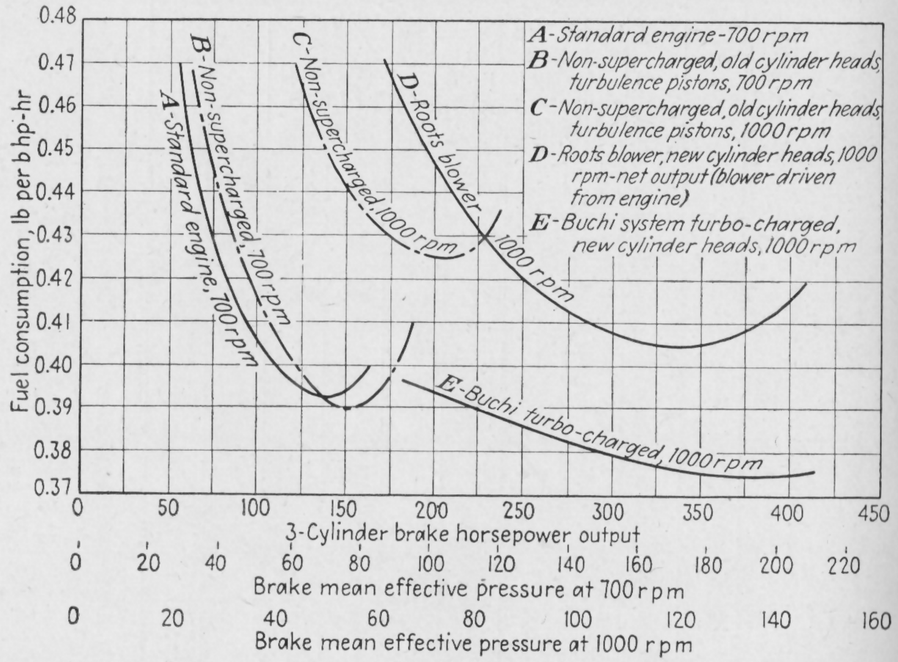

FIG. 209. Effect of various methods of supercharging on engine performance. (*Stewart, Boyer, and Anderson, Trans. SAE, Vol.* 49, 1941.)

by supercharging, the bsfc is likely to suffer if the supercharger is engine-driven. If the engine exhaust system is designed to deliver the exhaust gases to a turbine and the turbine is used to drive the supercharger, the ratio of bhp to ihp will increase as the amount of supercharging is increased if the engine-turbine-compressor combination is properly proportioned. A set of curves for the bhp and bsfc of an engine equipped with first an engine-driven and then an exhaust-turbine-driven supercharger are shown in Fig. 209.[6] The problems associated with the latter will be discussed further in the next chapter.

EFFECT OF ATMOSPHERIC CONDITIONS ON ENGINE OUTPUT

Atmospheric pressure, temperature, and humidity affect engine full-throttle output by influencing the air flow rate to the cylinders. Since the density of the air entering the cylinders of a spark-ignition

engine at full throttle is directly proportional to barometric pressure, the corresponding engine indicated output should be directly proportional to barometric pressure. This relation has been verified by many test data, including data for the wide variations in atmospheric pressure that are encountered with aircraft engines at high altitudes. The effects of inlet air temperature on engine output are much more complex. The largest single effect of temperature arises from its effect on air density. This tends to make the full-throttle indicated output inversely proportional to air temperature. However, as was pointed out in the section on Volumetric Efficiency (see Fig. 199), the smaller the temperature differential between the combustion-chamber walls and the induced charge, the less the heat transfer and hence the greater the volumetric efficiency. In practice, this effect operates in the opposite direction from the normal density effect but is roughly one-half as great. Test data on a wide variety of engine types have been analyzed and various empirical relations tried in an effort to find the one giving the best correlation.[1,8] These investigations have led to the general adoption of an inverse-square-root relation between power output and inlet air absolute temperature. Test codes usually specify that the correction be applied directly to the bhp, for this greatly simplifies the correction procedure. However, it gives a smaller correction than is justified.

Effect of Humidity. The water vapor in the air is incombustible and hence reduces the effective charge density. Allowance for this effect is usually made by taking the true barometer reading and subtracting the water-vapor pressure determined with a sling psychometer and a psychometric chart such as that of Fig. 210.

Power-correction Formulas. It is essential both for specifications and for development test work that full-throttle engine output be reducible to a common basis irrespective of the atmospheric conditions prevailing at the time of the test. The conditions chosen and in universal use are standard sea-level atmospheric conditions, viz. 29.92 in. Hg dry barometric pressure and 59.4°F (usually taken simply as 60°F). The term *dry barometer* means that the allowance has been made for humidity by deducting the water-vapor pressure from the true barometric pressure.

For most purposes the observed engine output may be corrected to standard conditions with sufficient accuracy by means of the following formula:[9]

$$\text{bhp (std cond)} = (\text{obs bhp})\left(\frac{29.92}{\text{obs dry barom}}\right)\sqrt{\frac{\text{obs abs temp}}{520}} \qquad (31)$$

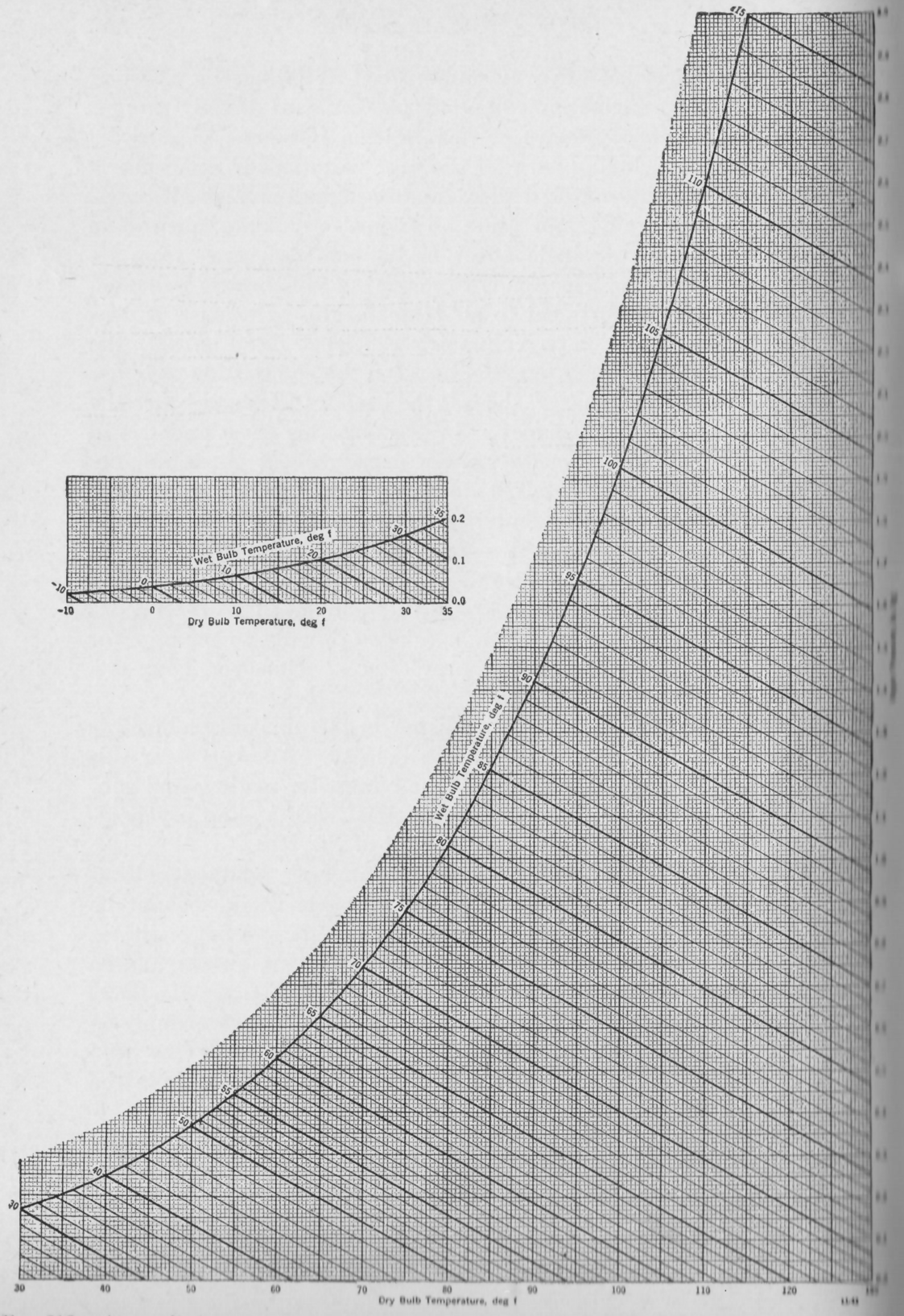

FIG. 210. Atmospheric water-vapor pressure as a function of dry- and wet-bulb thermometer readings. (*SAE Aircraft Engine Test Code.*)

A somewhat more elaborate expression may sometimes be justified for certain types of work in which the correction is to be placed on an indicated basis by including fhp, thus giving[9]

$$\text{bhp (std cond)} = (\text{obs bhp} + \text{fhp})\left(\frac{29.92}{\text{obs dry barom}}\right)\sqrt{\frac{\text{obs abs temp}}{520}} - \text{fhp} \tag{32}$$

Effect of Atmospheric Conditions on Specific Fuel Consumption. The large changes in power output brought about by changes in air density are not accompanied by corresponding changes in specific fuel consumption. As may be deduced from Fig. 212, the specific fuel consumption of a diesel is virtually unaffected by changes in atmospheric conditions, except in so far as they influence the fuel-air ratio. The same is true in a spark-ignition engine.

Power Corrections for Supercharged Engines. While the corrections discussed above were originally derived for unsupercharged spark-ignition engines, they are applicable to any other type with certain reservations. The characteristics of the supercharger affect the correction in highly supercharged spark-ignition engines because the operating point is inclined to shift its position on the supercharger performance curve and thus change the pressure ratio obtained at a given engine rpm. This effect is too complex to be discussed further here, but it can be visualized by inspecting Figs. 189 and 196.

Power Corrections for Diesel Engines. Diesel engines present a special problem in that the full-load output of a production engine is usually limited by the fuel-injection-pump setting. That setting is determined to allow the load to be increased only to the point where smoke and engine deposits begin to become objectionable. Since the air flow rate through a diesel engine depends on atmospheric pressure, temperature, and humidity in the same way as in a spark-ignition engine, the fuel-air ratio will increase with a reduction in inlet air density for a particular fuel-injection-pump setting. The resulting effects on engine output may be deduced from Figs. 211 and 212.[10] Figure 211 shows that, for any given set of inlet-air conditions giving an air density below the standard atmospheric value, as the fuel flow is increased a value is reached above which the bhp is definitely lower than the bhp at standard atmospheric conditions. From basic cycle-analysis considerations it follows that, if the ihp were plotted against fuel-air ratio, all the curves should be similar in shape and should peak at the same fuel-air ratio and that the peak output in each case should be directly proportional to fuel flow. That this would be the

case is demonstrated by Fig. 212, which shows the same data for a wide variety of inlet air densities replotted in terms of indicated horsepower-hours per pound of fuel vs. fuel-air ratio. All the data scatter around a common line showing that, at any given fuel-air ratio, the indicated horsepower-hours per pound of fuel is independent of air density. While the performance of some diesel engines may fall off with air density more rapidly than indicated by the above reasoning

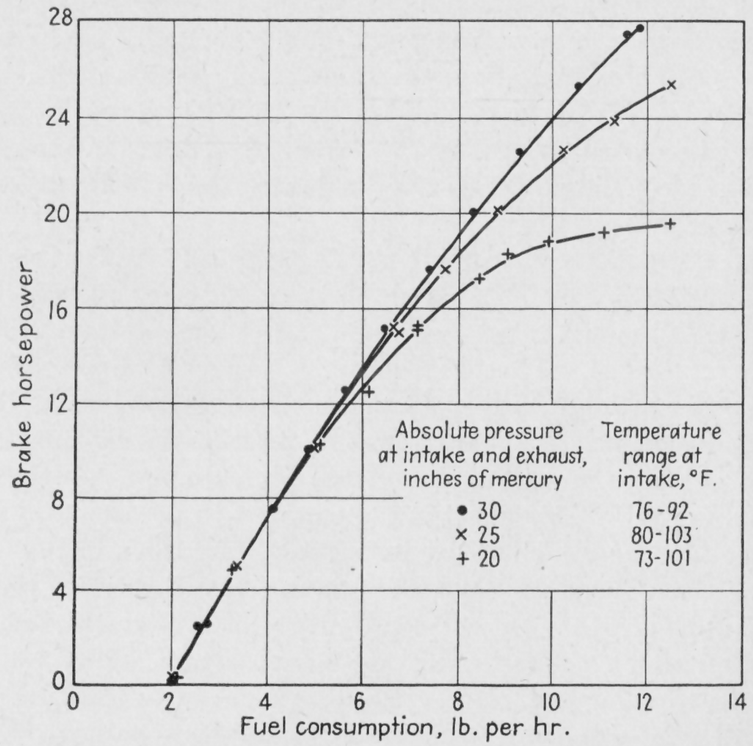

FIG. 211. Effect of ambient air pressure on the bhp of a diesel engine at 1,000 rpm (*Elliott, Trans. ASME, Vol.* 68, 1946.)

because of the dependency of spray-nozzle characteristics on air density (see Chap. VI), Figs. 211 and 212 represent the ideal that can be attained by many engines.

In applying the above relations to routine test work, it is evident that the small increase in the fuel-air ratio obtained with a reduction in air density at a fixed maximum fuel flow rate nearly offsets the effects on the bhp of the lower air flow rate for normal variations in barometric pressure. However, if the same engine were to be used in a truck for mountain operation at, say, 10,000 ft, the variation in fuel-air ratio would be large and the normal full-load fuel-air ratio would be greatly exceeded. For such conditions the best practice would

seem to be to use the same correction formulas as for spark-ignition engines while making allowances for the effects of changes in the fuel-air ratio.

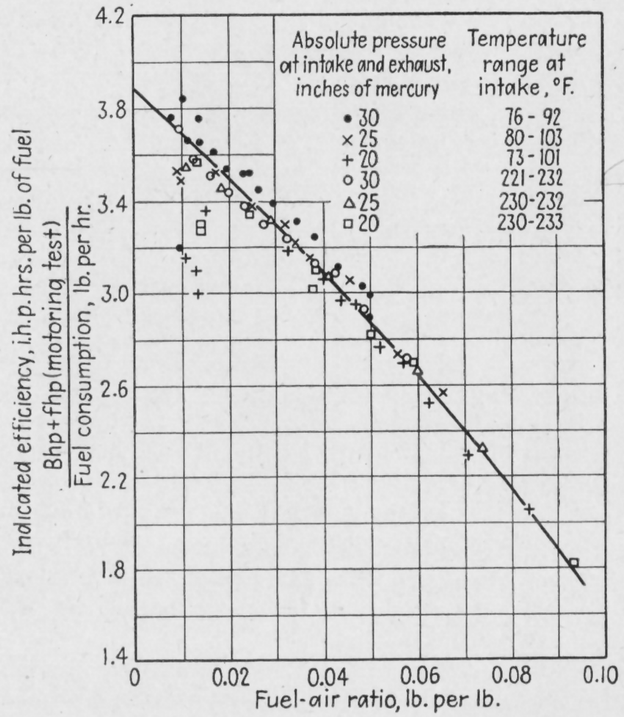

FIG. 212. Relation between indicated efficiency and fuel-air ratio for the data of Fig. 211. (*Elliott, Trans. ASME, Vol.* 68, 1946.)

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Problems

1. How would you expect the isac to vary in an idealized spark-ignition engine with (*a*) rpm, (*b*) load, (*c*) fuel-air ratio?

Sketch curves where appropriate.

2. Repeat the above for an actual engine. Indicate the magnitude (in percentage) of the deviations from the ideal to be expected in common automotive engines.

3. Would variations in the following quantities be likely to produce changes in the isac and, if so, what approximate percentage? Explain briefly in each case.

Spark advance	Fuel octane rating
Oil-pan temperature	Mixture distribution
Humidity	Cooling-water temperature

4. Explain briefly why a volumetric-efficiency curve might drop off at both low and high speeds in a given engine.

5. List three ways in which the fhp can be determined. State which is the most widely used and why.

6. Using the curves in Figs. 197, 198, 202, and 204, estimate the performance characteristics of an eight-cylinder 300-in.^{3}-displacement 7.2:1-compression-ratio spark-ignition engine. Assume a constant fuel-air ratio of 0.072. Prepare a curve for the full-throttle bhp and a set of curves showing bsfc vs. bhp for 500, 1,000, 2,000, 3,000, and 4,000 rpm.

7. Extend the above analysis for the full-throttle condition to include the effects of fuel-air ratio. Assume that the carburetor delivers a fuel-air ratio equal to that indicated by Fig. 98 at corresponding air flows. Allow for the difference in size between the two engines by taking the 140-cfm air-flow value in Fig. 98 as representing the same percentage of rated power output as the full-throttle 4,000-rpm condition for the engine of Prob. 6.

8. Justify the use of full-throttle-output corrections for atmospheric conditions as commonly applied to bhp, explaining in the process their relations to ihp.

9. If an engine delivers 77 bhp when the true barometer reads 29.46 in. Hg, the water-vapor pressure 0.37 in. Hg, and the room temperature 83°F, what would its output at the same speed be under standard atmospheric conditions?

10. State what correction if any should be applied to the bsfc in Prob. 9, and explain briefly.

CHAPTER XIV

GAS TURBINES

The rapid development of the gas turbine in recent years has led to talk of its use as a power plant for practically every conceivable application from aircraft, ships, locomotives, and central stations to motor vehicles. It happens that the gas turbine—like any other prime mover—has certain basic characteristics that make it exceptionally well suited to particular types of service. At the same time, these same or related characteristics may render it much less suited to certain other applications. Since the potential fields of piston- and turbine-type combustion engines overlap and since many basic problems are common to both types, it seemed that this text would not be complete without a chapter on the subject.

This chapter is divided into four sections. The first is concerned with the general layout of and problems associated with gas turbines and their components, the second with an analysis of the simple "open" gas-turbine cycle and the factors affecting it, the third with the performance characteristics of a typical gas turbine, and the fourth with the compounding of piston engines and gas turbines both for engine supercharging and as a means of increasing engine output and efficiency.

GAS-TURBINE FEATURES AND COMPONENTS

While the constant-pressure cycle (also called the *Joule* or *Brayton* cycle) was devised about the same time as the Otto and Diesel cycles, no commercial source of power was developed with it as a basis until the first units were used in petroleum refineries in 1936 (see page 225). Up to that time a commercially successful gas turbine was not possible because turbine-bucket materials capable of operation at sufficiently high temperatures to give good cycle efficiencies and suitable high-efficiency compressors were not available.[1,2] The preceding statement should be reread and underlined, for one is confronted at every turn in gas-turbine work with two major sets of problems—allowable turbine-bucket temperatures and compressor or turbine efficiency. These are almost invariably the determining factors in both design problems and performance limitations. The reasons for this

are forcibly demonstrated by an analysis of the idealized cycle when allowances are made for turbine and compressor efficiencies. While other factors such as combustion-chamber pressure drop and heat radiation also cause losses, their effects are small compared with those of compressor and turbine efficiency. In this connection it might be added that the gas turbine is much more susceptible to paper analysis than piston engines, so much so that a completely new model can be

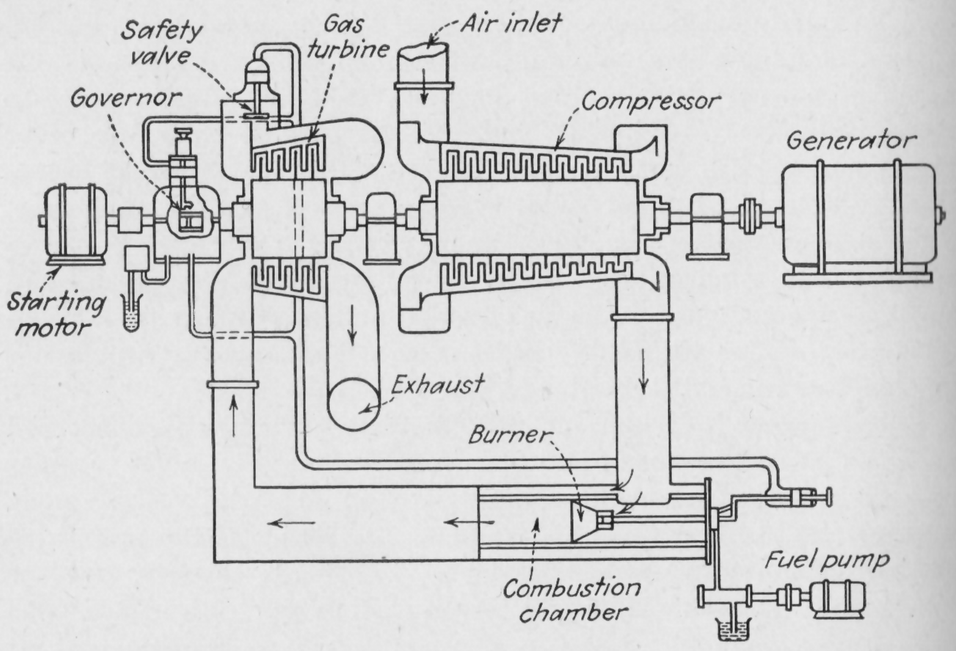


FIG. 213. Schematic diagram showing the principal components of a gas-turbine power plant. (*Meyer, Mech. Eng. Vol.* 61, 1939.)

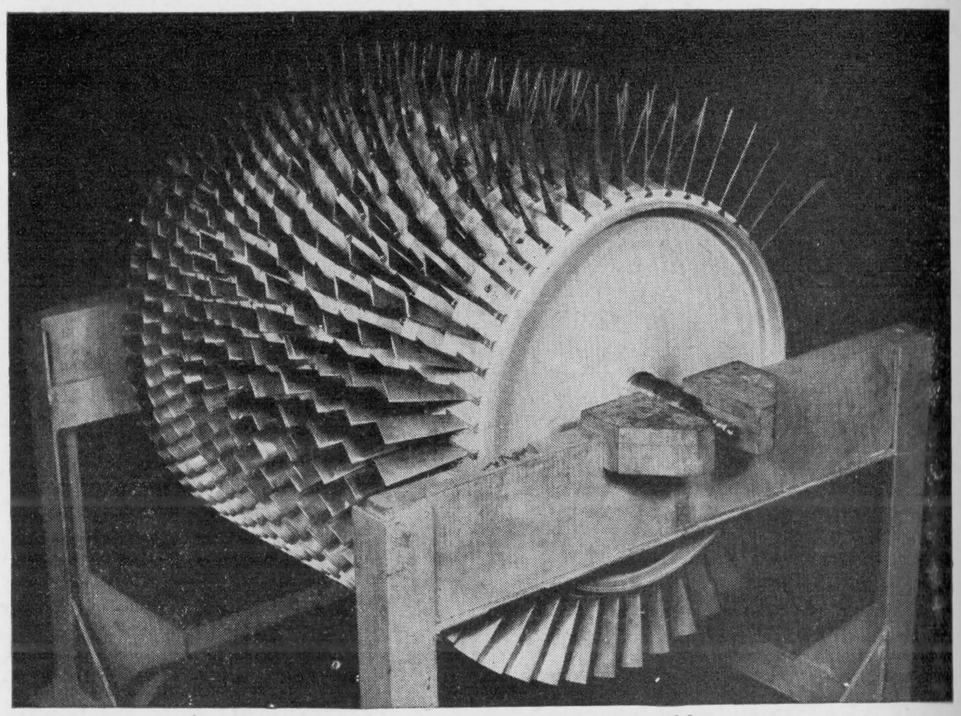
designed, built, and put into service in much less time than would be required for the development of a piston engine. (The greater mechanical complication of the latter makes a much greater amount of service testing necessary.) It should be emphasized, however, that a successful gas turbine can be designed only by the use of highly refined engineering techniques, particularly those related to aerodynamics.

Gas-turbine Components. A diagram showing the principal elements of a simple open-cycle type of gas turbine is shown in Fig. 213.[1] Air is compressed, fuel is sprayed into the air stream and burned at essentially constant pressure in a combustion chamber between the compressor and turbine, and then the gases expanded by the heat are allowed to escape through the turbine. If sufficient heat has been added in the combustion chamber and if the efficiencies of the turbine

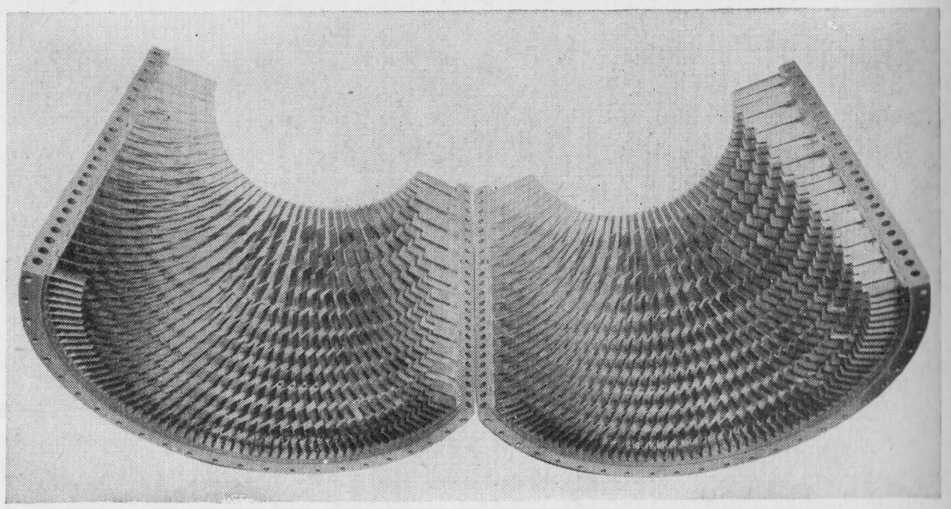
and compressor are sufficiently high, more power will be developed in the turbine than will be absorbed in the compressor so that the excess will be available for some useful purpose.

Types of Compressor Employed. The axial-flow compressor shown in Fig. 213 might be described as a multistage fan especially designed to give a high pressure rise per stage. The high pressure rise is accomplished by using a great many small, closely spaced airfoils driven at a high speed (see Fig. 214). Other types of compressor such as those described in Chap. XII, Supercharging, are also employed. Roots blowers are not well suited to gas-turbine use because of their relatively poor efficiency at the high pressure ratios necessary, but centrifugal and Lysholm compressors having much the same appearance as the superchargers in Figs. 190 and 194 are being used quite successfully.[3] It should be mentioned that the performance characteristics of an axial-flow compressor are very similar to those of a centrifugal unit (see Fig. 196), with the exception that the constant-speed curves for pressure ratio vs. flow do not extend over so great a range of flow in comparable machines. This makes the efficiency of the axial-flow compressor much more sensitive to air flow rate. Another inherent characteristic of axial-flow compressors designed for high-speed operation is poor efficiency at low speeds. This is the reason why gas turbines employing axial-flow compressors require starting motors many times larger than comparable units employing centrifugal compressors.

Combustion Chambers and Burners. The theoretical and practical aerodynamic problems associated with the development of high-efficiency compressors and turbines were so great that prior to the actual construction and testing of the first successful unit relatively little concern was felt over a set of parts that appear as simple as a burner and a combustion chamber. Yet these very parts almost completely halted development of the first aircraft turbojet engines in spite of the efforts of some of the best British and American combustion engineers during the critical war years of 1939 to 1941. The main difficulty was that the flame would "blow out" at irregular intervals for no apparent reason. The problems involved were quite different from those with respect to any previous type of burner because of the high air velocities necessary (300 to 400 ft/sec) to keep the combustion-chamber size within reasonable limits and the low fuel-air ratios of 0.005 to 0.020, which are much lower than the lean combustibility limit.[4] A satisfactory solution to the gas-turbine combustion-chamber problem was evolved by separating the air stream into



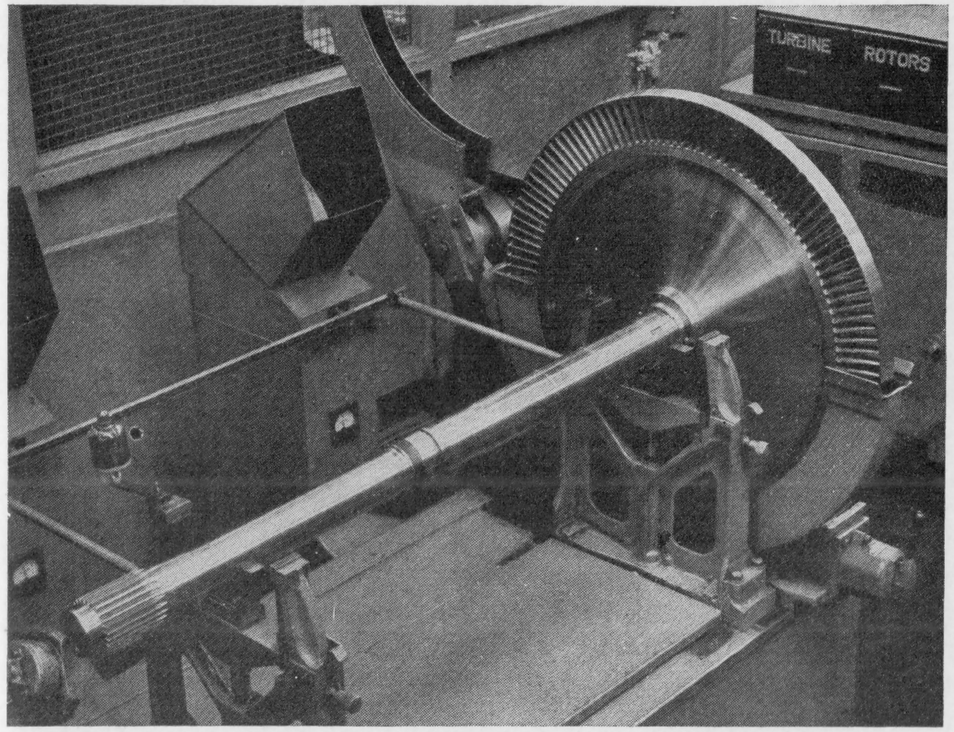
(*a*) Axial-flow compressor rotor assembly.



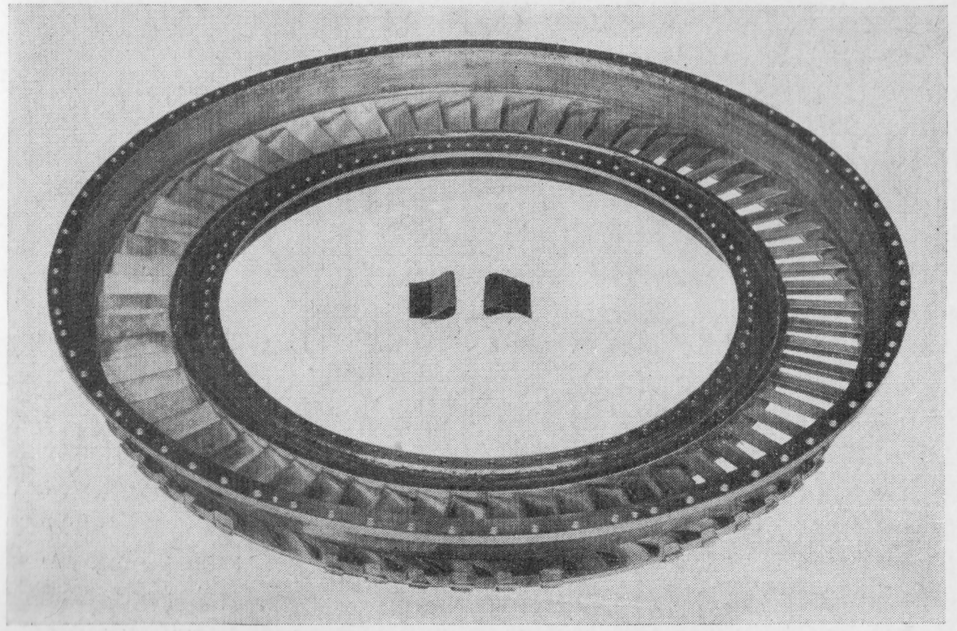
(*b*) Compressor stator assembly.

FIG. 214. For descriptive legend see opposite page.

two parts and burning the fuel with only one portion—the primary air—slowed down to a reasonably low velocity and then allowing the remaining unburned fuel vapor and the combustion products to mix gradually with the rest of the air. This arrangement permits most of the combustion process to take place under rich mixture conditions.



(*c*) Turbine wheel and shaft assembly.



(*d*) Turbine nozzle diaphragm assembly.

FIG. 214. Principal parts of a turbo-jet aircraft engine. (*Thompson Products, Inc.*)

After combustion is nearly complete the gases can begin to mix so that they reach the turbine at an essentially uniform temperature.[5,6] The high air velocities in gas-turbine combustion chambers have made possible extremely high heat release rates. Values for subsequently developed units have run well over 20,000,000 Btu/hr per cubic foot of combustion-chamber space as compared with only about 1,000 Btu/(hr)(ft^3) for the highest heat release rates in boiler furnaces. While their combustion chambers operate under very different conditions, spark-ignition and diesel engines at full load commonly release 60,000,000 Btu/(hr)(ft^3) of combustion-chamber space, giving values of the same order of magnitude as gas-turbine combustion chambers.

Turbine Blading. Gas-turbine wheels are made in much the same way as those for steam turbines except that the ratio of blade length to wheel diameter is greater for the gas turbine. Three types of blading may be used—impulse blading, reaction blading, or a combination of the two. Reaction blading can be made to give the best blade efficiency, but other factors must also be considered. The impulse type of turbine is lighter and more compact, for a pressure ratio of 2:1 can be used across a single stage with good efficiency, whereas a pressure ratio of less than 1.3:1 per stage must be employed with reaction blading. Another factor favoring the impulse type is that, since only a small part of the gas expansion takes place ahead of the first stage of reaction blades, the gases strike the blades at a temperature only a little below the turbine inlet temperature. If an impulse type of turbine is used, the gas can be expanded in a nozzle ahead of the turbine buckets so that the latter will operate at a temperature somewhat below that of the gas at the turbine inlet. This permits a higher turbine inlet temperature and hence both a higher over-all output and a higher efficiency.

The very great effects of allowable turbine inlet gas temperatures on both the output and the efficiency of any given unit have been mentioned and will be clearly demonstrated a little later in the chapter. Allowable gas temperatures can be increased in two ways—by increasing the allowable operating temperature of the turbine buckets through the use of special materials and by cooling the buckets so that their temperature is below that of the gas stream. Since any alloy has a lower melting point than that of the highest-melting-point constituent element and since ceramic and crystalline materials are inherently brittle, it seems unlikely that any spectacular improvements in gas-turbine performance will be brought about suddenly by a new material. The turbine-bucket materials in use at the time of writing are in most cases alloys of such metals as cobalt, chromium, tungsten, nickel, molybdenum, and columbium. In turning to the other alternative of blade cooling, one finds that, while some turbine-bucket cooling will always take place because of heat transfer from the base of each bucket to the turbine wheel and thence to the shaft, the amount can be increased by purposeful design. Air or oil cooling of either the wheel or the buckets themselves may be employed. One of the most effective procedures is to bleed off air from the compressor and feed it through the turbine shaft and wheel to hollow buckets. If this is done, the cooling air is usually discharged radially from the tips of the buckets and allowed to mix with the main gas stream. While aerodynamic losses associated with such an arrangement act to offset somewhat the possible improvements in cycle efficiency from higher allowable gas temperatures, a great increase in output from a given unit can be obtained because so much more heat may be added to the air stream passing through the combustion chamber.

Regeneration, Intercooling, and Closed Cycles. The output and efficiency of a gas-turbine cycle may be improved in a number of ways. A part of the heat in the gases leaving the turbine may be utilized to heat the air leaving the compressor before it reaches the combustion chamber and thus reduce the amount of fuel required.[7] This is called a *regenerative cycle*. The size and weight of the heat exchanger required for appreciable benefits from this source render it of questionable value for vehicle-power-plant installations. The pressure drop through the heat exchanger as well as the cost and the space required for both the heat exchanger and the necessary extra duct work are other disadvantages. These disadvantages are, of course, much more than offset by the higher efficiency for stationary-power-plant installations.

The power required for compression of the air can be reduced by intercooling it between stages. This gives a slight increase in cycle efficiency. It serves to give a denser medium leaving the compressor, thus reducing the size of the combustion chamber and turbine and hence their cost and space requirements. On the other hand, the cost, weight, and space necessary for the intercooler operate in the opposite direction.

All the discussion up to this point has been concerned with the "open-cycle" gas turbine, in which any given element of air passes through the turbine only once and then returns to the atmosphere. Closed-cycle gas turbines have been built for stationary power plants.[8]

In units of this type, air leaves the turbine under a pressure of several atmospheres, is cooled in heat exchangers, enters the compressor and is compressed, then is heated by passage through another set of heat exchangers and returned to the turbine. One set of heat exchangers is cooled by ambient air or water, while the other set is heated by a source of energy such as a furnace or a nuclear reactor. While this cycle requires heavy and bulky heat exchangers, the greater air densities obtained by pressurizing the system permit a smaller sized compressor and turbine for a given power output. It is also practically free of trouble from deposits on turbine and compressor blades and the losses in efficiency that such deposits cause.

In reviewing the above and summarizing, it appears that the simple open-cycle type of gas turbine is better suited than the others to automotive and aircraft equipment because of the cost, weight, and space requirements of heat exchangers.

EFFICIENCY OF ACTUAL GAS-TURBINE CYCLES

As was found to be the case with the constant-volume cycle in Chap. V, it is necessary to include the effects of differences between the ideal and the actual constant-pressure cycle both to obtain a good approximation to the output and fuel consumption of an actual unit and to evaluate the relative effects of each of the operating variables.

Factors Determining Cycle Efficiency. In Chap. II, Idealized Cycles, it was demonstrated that the efficiency of the ideal cycle is dependent only on the pressure ratio employed and the thermodynamic properties of the working fluid. These quantities, therefore, must be expected to have important effects on the actual cycle.

By far the greatest sources of difference between the ideal and the actual cycle in a gas turbine are the aerodynamic, or fluid-friction, losses in the compressor and turbine. Similarly, by far the most important operating variables determining output and efficiency are the air inlet and turbine-bucket temperatures. While other factors may influence output and/or efficiency, their effects are relatively very much smaller.

The pressure-volume and temperature-entropy diagrams for both an ideal and an actual cycle as shown in Figs. 215 and 216 help one to visualize the qualitative effects of these losses and limitations. The aerodynamic losses cause the compression and expansion processes to differ considerably from the ideal isentropics for which the entropy would stay constant. Note particularly the manner in which these compression and expansion lines on the temperature-entropy

diagram depart from the vertical. A higher allowable peak temperature in the cycle would permit the expansion line to be moved to the right on the pressure-volume diagram and hence permit an increase in diagram area, *i.e.*, in cycle output. A reduction in air-inlet temperature would cause the compression curve to be moved to the left and thus also yield an increase in cycle output.

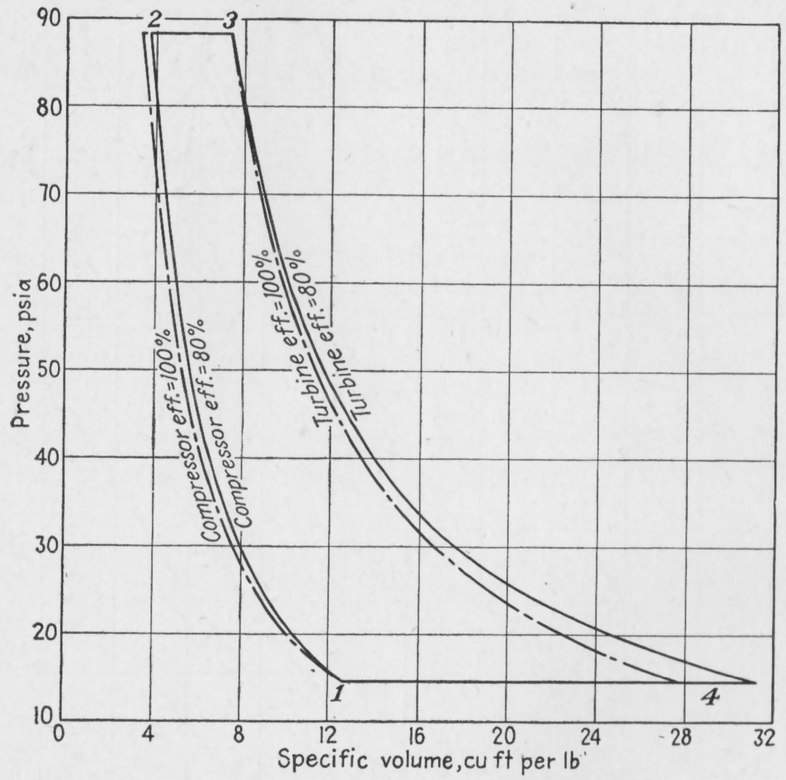


FIG. 215. Pressure-volume diagram for a 6.0:1 pressure ratio gas-turbine cycle with a turbine inlet temperature of 1300°F.

Probably the simplest way to obtain a quantitative relationship between the factors limiting output or causing losses in thermal efficiency is to write an expression for the net work output from the cycle divided by the heat input and then, by algebraic manipulation, obtain one or more expressions that may be used with reasonable convenience to determine the effects of such individual quantities as compressor efficiency and turbine-bucket temperature. On this basis we may write

$$\text{Thermal eff} = \frac{(\text{work from expansion}) - (\text{work of compression})}{(\text{heat-energy input})} \qquad (33)$$

The above expression can be made more explicit by substituting

several known relations. For example, the heat-energy input in the combustion chamber in foot-pounds is given by

$$\text{Energy into cycle} = 778Wc_p(T_3 - T_2)$$

where the numerical subscripts refer to the corresponding points in the P-V diagram for the cycle in Fig. 215.

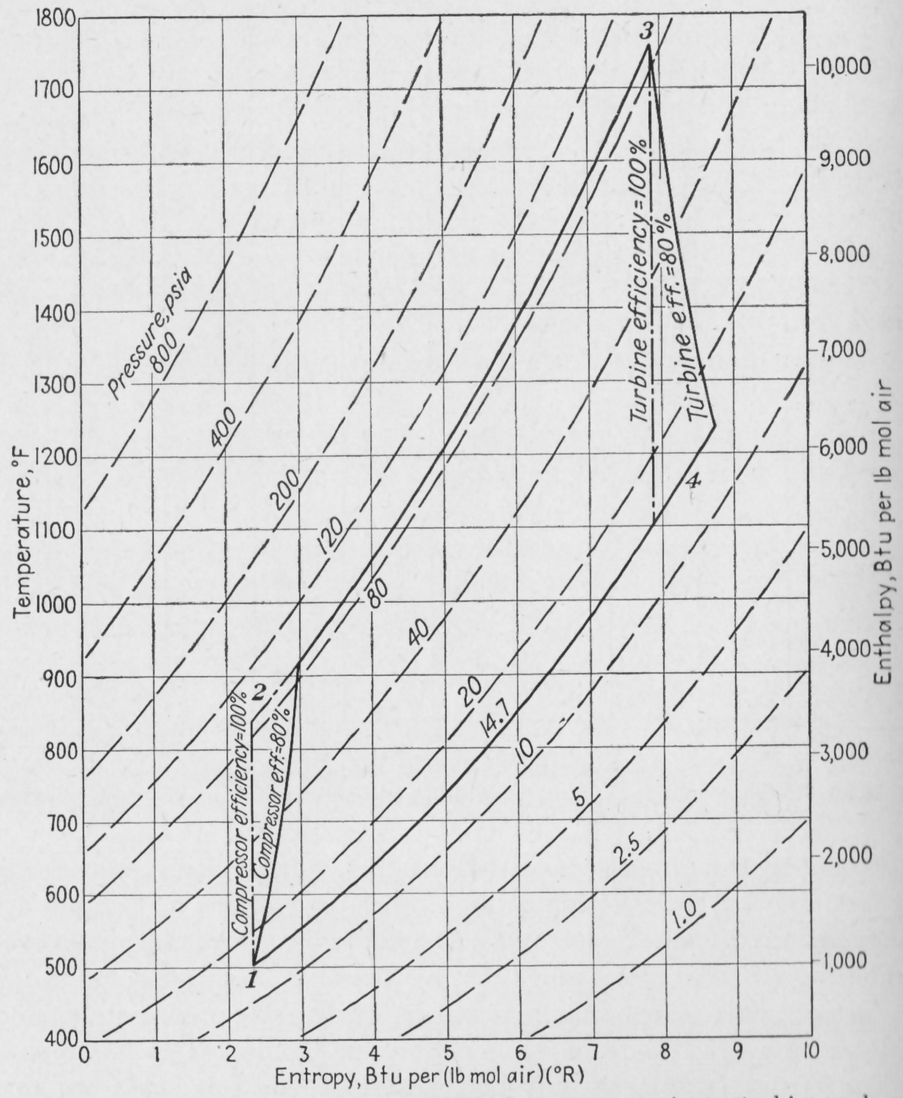


FIG. 216. Temperature-entropy diagram for a 6:1 pressure ratio gas-turbine cycle with a turbine inlet temperature of 1300°F.

The actual power input to the compressor can be expressed as a function of the compressor efficiency and the power for an isentropic compression. Since an actual compressor of the type employed in

gas turbines handles such large quantities of air at such high flow rates, there is very little heat transferred from the air during the compression and the power required for an isentropic compression represents the minimum ideally obtainable. The compressor efficiency has, therefore, been defined as

$$\eta_c = \frac{\text{isentropic work}}{\text{actual work}} \tag{34}$$

Thus the actual work required for compression is

$$\text{Actual compressor work} = \frac{k/(k-1)WRT_1[(P_2/P_1)^{\frac{k-1}{k}} - 1]}{\eta_c}$$

This same concept was developed in Chap. XII, Supercharging, and hence needs no elaboration. The efficiency of a turbine has been defined in much the same way as the corresponding term for the compressor except that less work is always obtained from the actual turbine than from an ideal isentropic expansion. Eddies and fluid friction in the gas stream as it passes through the turbine nozzles and blades are the main source of loss. Thus the turbine efficiency has been defined as

$$\eta_t = \frac{\text{actual work}}{\text{isentropic work}} \tag{35}$$

The actual-work input from the gas to the turbine wheel is then

$$\text{Actual turbine work} = \eta_t \left\{ \frac{k}{k-1} WRT_1 \left[1 - \left(\frac{P_4}{P_3} \right)^{\frac{k-1}{k}} \right] \right\}$$

If the pressure drop between the compressor and turbine is neglected, the terms involving the pressure ratio in the above expressions may be related as follows:

$$\frac{P_2}{P_1} = \frac{P_3}{P_4}$$

Hence

$$\left(\frac{P_2}{P_1} \right)^{\frac{k-1}{k}} = \frac{1}{(P_4/P_3)^{\frac{k-1}{k}}}$$

Since the quantity Y for the compressor has been defined on page 324 as

$$Y = \left[\left(\frac{P_2}{P_1} \right)^{\frac{k-1}{k}} - 1 \right]$$

then

$$\left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}} = Y + 1 = \frac{1}{(P_4/P_3)^{\frac{k-1}{k}}}$$

or

$$1 - \left(\frac{P_4}{P_3}\right)^{\frac{k-1}{k}} = 1 - \frac{1}{Y+1} = \frac{Y}{Y+1}$$

The various values thus defined for turbine work output, compressor work, and heat input may be substituted in Eq. (33) to give

$$\text{Thermal eff} = \frac{\dfrac{kWRT_3Y\eta_t}{(k-1)(Y+1)} - \dfrac{kWRT_1Y}{(k-1)\eta_c}}{778Wc_p(T_3 - T_2)}$$

By definition of compressor efficiency (see Eq. (26), page 324),

$$T_2 = T_1\frac{Y}{\eta_c} + T_1$$

Combining terms and substituting for T_2 gives

$$\text{Thermal eff} = \frac{\dfrac{k}{k-1}WR\left(\dfrac{T_3\eta_tY}{Y+1} - \dfrac{T_1Y}{\eta_c}\right)}{778Wc_p\left(T_3 - T_1\dfrac{Y}{\eta_c} + T_1\right)} \tag{36}$$

But

$$R = (c_p - c_v)778 \qquad k = \frac{c_p}{c_v}$$

Combining the constants in the numerator and denominator of Eq. (36) gives

$$\frac{Rk}{778c_p(k-1)} = \frac{778(c_p - c_v)\dfrac{c_p}{c_v}}{778c_p\left(\dfrac{c_p}{c_v} - 1\right)} = \frac{\dfrac{c_p}{c_v} - 1}{\dfrac{c_p}{c_v} - 1} = 1$$

Hence

$$\text{Thermal eff} = \frac{\dfrac{T_3\eta_tY}{Y+1} - T_1\dfrac{Y}{\eta_c}}{T_3 - T_1\dfrac{Y}{\eta_c} + T_1} \tag{37}$$

Inspection of Eq. (37) discloses that the first term of the numerator is the temperature drop in the turbine, the second is the temperature

rise in the compressor, and the denominator is the temperature rise in the combustion chamber. Since the energies involved in each of these three stages of the cycle are directly proportional to the temperature changes, this gives a good simple check on the derivation.

Inspection of Eq. (37) will show that the efficiency of an actual gas-turbine cycle is heavily dependent on five quantities—the pressure

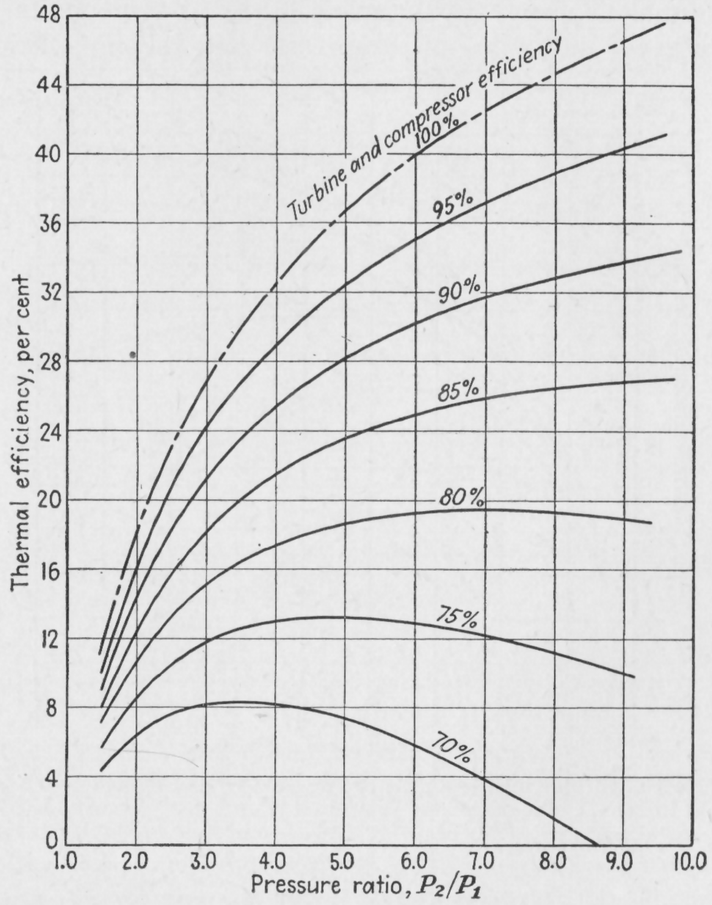


FIG. 217. Effect of turbine and compressor efficiencies on the thermal efficiency of a simple open cycle for a gas turbine. The data were computed from Eq. (36) on the basis of a compressor air-inlet temperature of 60°F and a turbine inlet temperature of 1500°F.

ratio (which determines the Y factor), the air inlet temperature, the compressor efficiency, the turbine inlet temperature, and the turbine efficiency. The individual effects of each of these will be discussed in the following paragraphs.

Effects of Pressure Ratio and Turbine and Compressor Efficiencies. An interesting demonstration of the effect of turbine and compressor efficiency on the thermal efficiency of the constant-pressure cycle for a given set of temperature limits is shown in Fig. 217. The

compressor and turbine efficiencies were assumed equal for each set of conditions because the degree of aerodynamic refinement in any given power plant is usually substantially the same in both the compressor and the turbine. The dotted line for the ideal cycle shows the limiting thermal efficiency of the cycle for 100 per cent efficiencies in the compressor and turbine. An important point is that the curve for the thermal efficiency of the cycle peaks at some finite value of pressure ratio for any value of compressor and turbine efficiency less

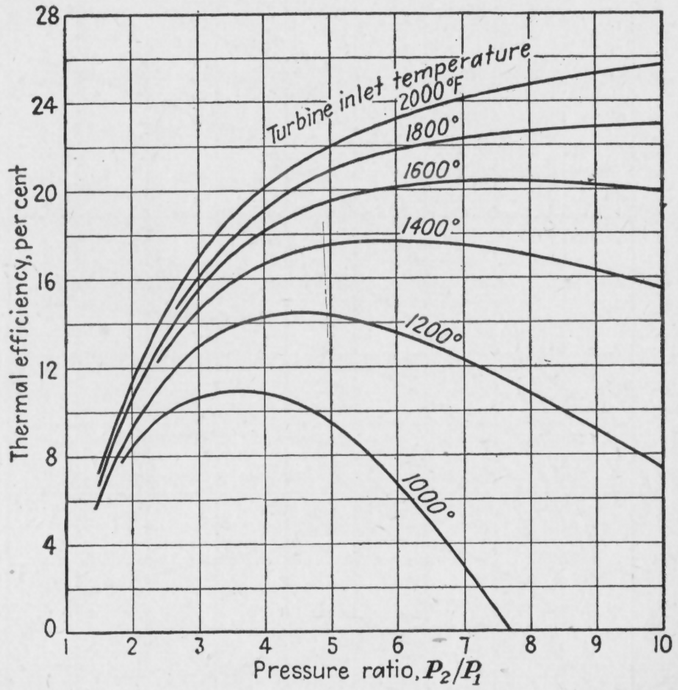


FIG. 218. Effect of turbine inlet temperature on the thermal efficiency of a simple open cycle for a gas turbine. The data were computed from Eq. (36) on the basis of turbine and compressor efficiencies of 80 per cent and a compressor air-inlet temperature of 60°F.

than 100 per cent. Note that the lower the compressor and turbine efficiencies the lower the pressure ratio at which the peak of the thermal efficiency curve occurs.

Effects of Turbine-inlet Temperature. Figure 218 shows how very rapidly the thermal efficiency of a constant-pressure cycle increases with turbine inlet temperature and demonstrates the importance of the problem of high-temperature-resistant turbine-bucket materials in the gas-turbine field.

Effects of Compressor-inlet Temperature. The effect of the air temperature at the compressor inlet on the thermal efficiency of the cycle is shown in Fig. 219 for a representative set of values for turbine-

inlet temperature, pressure ratio, and compressor and turbine efficiencies. Since air temperatures run around −65°F at high altitudes, this curve indicates one of the reasons why the gas turbine is especially well suited to aircraft work.

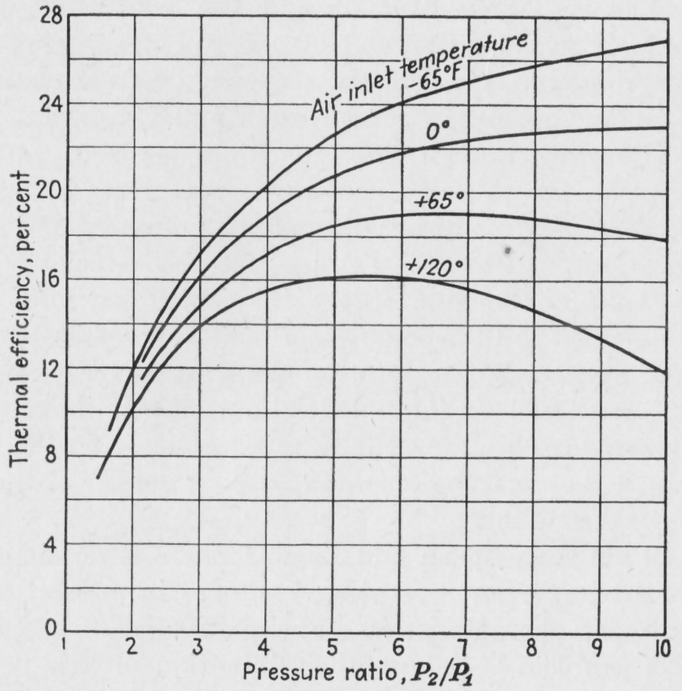


FIG. 219. Effect of compressor air-inlet temperature on the thermal efficiency of a simple open cycle for a gas turbine. The data were computed from Eq. (36) for a constant turbine inlet temperature of 1500°F and turbine and compressor efficiencies of 80 per cent.

Variable Turbine and Compressor Efficiencies. The data given in Figs. 217 to 219 are for gas-turbine *cycles*, not gas-turbine *power plants*. In the latter, such important factors as turbine and compressor efficiency vary considerably with changes in speed and gas flow rate (see Fig. 196), and hence compressor and turbine efficiencies cannot be held constant in any given power plant.

GAS-TURBINE POWER-PLANT-PERFORMANCE CHARACTERISTICS

The presentation of the performance characteristics for even the simple open-cycle type of gas-turbine power plant is very complex, partly because of the range of possible pressure (or compression) ratios and partly because the differences between the ideal and the actual cycle are not only large but vary widely with the prime operating conditions, speed and load. As shown in Chaps. V and XIII, this is not true in piston-type internal-combustion engines, for in that type

the indicated thermal efficiency is affected only to a minor degree by practically all operating variables in the normal operating range for either spark-ignition or diesel engines.

Effects of Speed on Output and Efficiency. A number of factors not included in the above discussion on the gas-turbine cycle have important effects on the output and efficiency of an actual power plant. Probably the most important of these are the inherent characteristics of centrifugal and axial-flow compressors and turbines. The volumetric air flow rate through these machines tends to vary directly with their speed, the pressure rise in the compressor tends to vary roughly as the square of the speed, and the power input to the compressor or the output from the turbine (for good efficiency) tends to vary as the cube of the rpm. Since the centrifugal force acting on the turbine buckets and the compressor blades increases as the square of the speed, stresses in the rotating elements increase very rapidly with speed. Inasmuch as the potential output also increases rapidly with speed, gas turbines are usually rated at close to the maximum speed at which they may be operated without danger of turbine- and compressor-bucket failures.

Relations between Speed and Load. There is no simple way of relating all the important operating variables in presenting gas-turbine performance characteristics. Figure 220 shows one method of handling the problem.[9] A careful examination of this performance chart will prove very instructive. Note how rapidly both the capacity and the pressure ratio of the machine fall off as the speed is decreased. Note, too, that the lines of constant output indicate that the speed for minimum fuel consumption drops off somewhat as the load is decreased. The lines of constant turbine inlet temperature indicate the relationship of that quantity to load and fuel flow rate.

Two very interesting curves are superimposed on the other networks. One is a curve for a cubic relationship between load and speed, *i.e.*, a good approximation for the steady-speed power requirements of a motor vehicle, a train, airplane, or ship. The other is for a constant turbine-bucket temperature. As might be expected from the cubic speed-load characteristic of the turbine and compressor as mentioned in the previous paragraph, these curves are almost identical. From them it is evident that the gas turbine is very well suited to locomotive, marine, or aircraft installations but much less well suited to motor-vehicle use. The reason for this is that the propeller load in a ship or an airplane follows a cubic curve rather closely, and there is relatively little variation in the torque required at any given

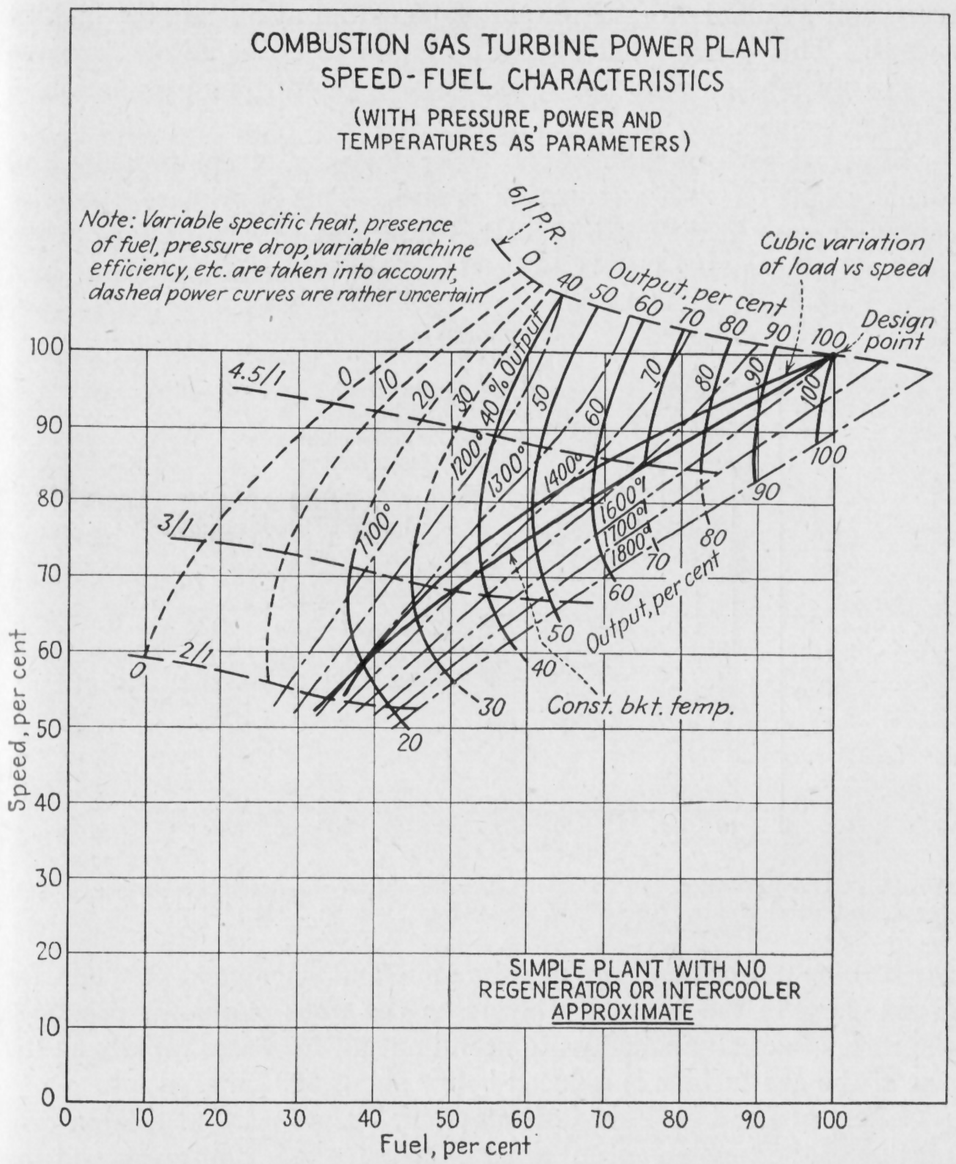


FIG. 220. Performance characteristics of a gas-turbine power plant. The dash-dot lines are for constant values of turbine inlet temperature in °F, while the long dashed lines are for constant values of pressure ratio. (*Salisbury, Mech. Eng., Vol.* 66, 1944.)

rpm. In a locomotive, while the load deviates widely from a cubic curve in starting or hill climbing, the gas turbine can be used to drive an electric generator and hence can be operated at top speed while the train is starting up—the generator and driving motors serving as an infinitely variable speed transmission. While the road-load curve for a motor vehicle is very close to the cubic characteristic of the gas turbine, acceleration requirements and steep grades make a large

reserve of available torque absolutely essential at all but the highest speeds. This means that something close to an infinitely variable speed transmission would be required for a gas-turbine installation in a motor vehicle.

Effect of Size on Efficiency. The efficiency of compressors and turbines falls off with a reduction in size. This is primarily because of the increase in the ratio of wetted wall area to flow-passage cross-sectional area as the size is reduced. Since the principal losses in a

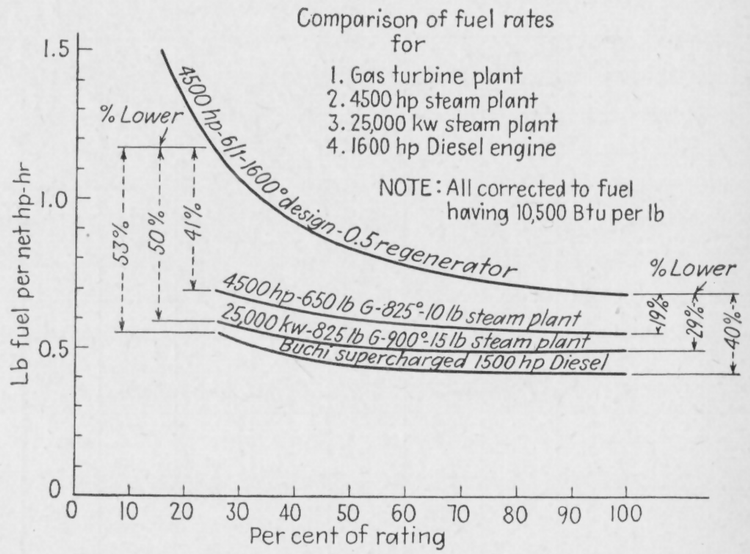


FIG. 221. Comparison of the specific fuel-consumption characteristics of gas turbine, steam, and diesel plants. (*Salisbury, Mech. Eng., Vol.* 66, 1944.)

gas-turbine power plant are aerodynamic and since most of the latter losses occur in the boundary layers on the walls of the air passages, the ratio of aerodynamic loss to useful output increases rapidly as the size of the gas turbine is reduced below about 500 bhp output.

Effect of Load on Fuel Consumption. The fact that a large proportion of the turbine output must go to drive the compressor and the fact that the air flow rate through an axial or centrifugal compressor cannot be varied appreciably without changing the speed (and hence the pressure ratio) act to give gas-turbine power plants poor fuel economy for part-load operation. The nature of this effect on several types of prime mover suited to stationary electric-power generation is shown in Fig. 221. It should be mentioned, though, that the initial cost of a gas-turbine plant would be much less than that for any of the other types, and hence the overhead for a gas-turbine stand-by power plant might be sufficiently lower to justify its use.

Specific Air Consumption. The usefulness and importance of specific air consumption in reciprocating-engine work were discussed in the previous chapter. The large amounts of air handled and the size of the necessary ducts for gas turbines make the same parameter also useful in this field. As indicated in Fig. 222, the specific air consump-

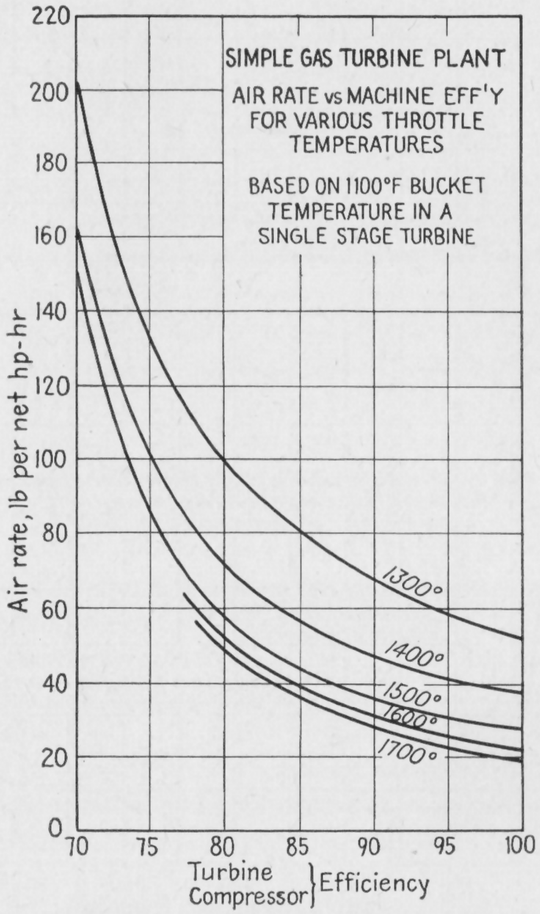


FIG. 222. Brake specific air consumption of a simple open-cycle gas-turbine power plant as a function of turbine and compressor efficiency and turbine inlet (throttle) temperature. (*Salisbury, Mech. Eng., Vol.* 66, 1944.)

tion of a typical gas turbine might run from 20 to 200 lb/bhp-hr, or three to thirty times the corresponding values for a spark-ignition engine.

COMPOUNDING OF PISTON ENGINES AND GAS TURBINES

The concept of using a turbine to recover some of the energy left in the gases in a constant-volume cycle at the end of the power stroke of the piston was originated as the gasoline automobile began to come into use.[10] As was the case with other forms of the gas turbine, how-

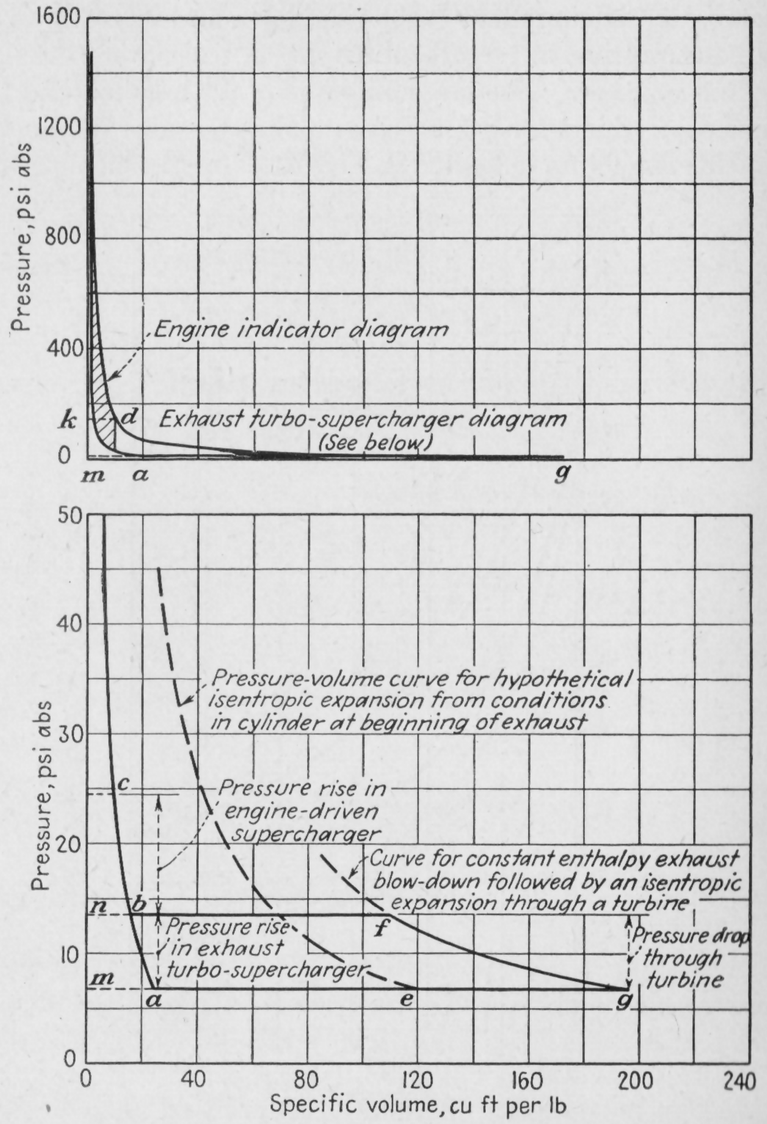


FIG. 223. Pressure-volume diagrams for an aircraft engine for operation at 20,000 ft altitude using an exhaust turbine-driven supercharger in conjunction with a supercharger in the engine driven from the crankshaft. The lower diagram shows the low-pressure portion of the upper diagram with the vertical scale magnified forty times.

ever, it could not be developed properly until high compressor and turbine efficiencies could be attained and high-temperature-resistant alloys became available. Turbine compounding of piston engines has since become practicable, and many applications have been made in certain fields since the early 1930's.

The amount of power ideally available from the exhaust of a piston engine and possible methods of obtaining it can be visualized by

examining Fig. 223. In the upper diagram the air compression in the supercharger and the exhaust-gas expansion through the turbine are shown to the same scale as the events in the cylinder to give some idea of the relative areas of the various portions of the compound cycle. The low-pressure portion of this complete diagram has been replotted beneath it and the vertical scale magnified forty times to resolve some of the details. The power ideally available from the exhaust gas is proportional to the area *demk*. It is not practicable to obtain all this power from an actual machine, however, for the process in the piston engine is inherently intermittent. Only the first portion of the exhaust gas to leave the cylinder could enter a turbine at high pressure—the balance would have to flow out at progressively lower pressures until the pressure dropped to some low equilibrium value. An intermittent flow of this sort cannot be handled well in a turbine, for the efficiency of the blading is so sensitive to gas velocity that good efficiency can be had for only a rather narrow range of pressure differential across the turbine nozzles and blades. Inasmuch as this sensitivity to gas flow rate increases rapidly with the number of turbine stages and since the highest pressure ratio per stage for good efficiency is only about 2.0:1, it follows that it is very difficult to obtain more than a moderate amount of the energy ideally available in the exhaust blowdown. Two types of arrangement have been employed. In one, the exhaust gas is discharged into a manifold that acts as a surge chamber between the engine and the turbine and dampens out the individual exhaust impulses. The pressure in the manifold is maintained above atmospheric so that the gas can be expanded through the turbine and power proportional to area *fgmn* obtained. If the turbine is used to drive a supercharger as in the example in Fig. 223, the net output from an ideal compressor-turbine unit is then proportional to the area *abfg*. Figure 223 was prepared for a single-stage unit; but, if desired, more back pressure could be imposed on the engine, and more stages could be added to the turbine. Increasing the engine exhaust back pressure has the disadvantage that it increases the exhaust-gas temperature, which is inclined to cause trouble with both the engine exhaust valves and the buckets in the first stage of the turbine. It also reduces engine volumetric efficiency, as shown in Fig. 200.

Note that the expansion line for the turbine in Fig. 223 is parallel to but above an extension of the isentropic for the piston-engine portion of the cycle. This is caused by the fact that all the kinetic energy built up in the exhaust-blowdown process is dissipated in eddies

so that it appears as heat in the exhaust gas entering the turbine. Thus the exhaust-blowdown event is a constant-enthalpy process, and an increase in entropy occurs.

Performance Characteristics. It has been found that the very dissimilar performance characteristics of piston engines and turbines make the problems of control very complex. While this would not be the case if the power-plant application were such that the machine would always run at a given speed and load, there are very few such applications. Even in stationary diesel engines, where both the engine speed and air flow rate remain substantially constant, the variation in exhaust-gas temperature and velocity with load is such that good turbine performance would be obtainable only over a portion of the load range unless a variable-speed transmission and suitable control mechanism could be provided. This would be complicated by such problems as a means of distributing the load between the engine and turbine. In a spark-ignition-engine application—in which the intake manifold pressure and the air flow rate vary widely with both speed and load—the problems become far more complex. Turbine- and piston-engine performance characteristics do not match, for the full-load output of the former varies as the cube of the speed, while that of the latter varies directly as the speed. Because of this and because of problems of control, the cost and complexity of the required equipment, etc., there are only two types of application of compounded piston-turbine engines in fairly widespread use at the time of writing. In both cases the turbine is used to drive a supercharger, and no attempt is made to obtain power from the turbine other than that required for supercharging. Thus in both cases the function of the turbine is simply to increase the engine output by supercharging, and there is no gain in the over-all thermal efficiency of the power plant.

Exhaust Turbosuperchargers for Aircraft Engines. The exceptionally good high-altitude performance characteristics of aircraft engines equipped with exhaust turbosuperchargers have been well publicized in connection with military aircraft. The engine exhaust manifold is conventional except that all the exhaust gases may be directed to a turbine, or a *waste gate* may be opened so that a part of them by-passes the turbine and goes directly to the atmosphere. No effort is made to recover the energy in the exhaust blowdown because it is not needed. In fact, as demonstrated in Fig. 223, even with relatively inefficient turbines there is usually an excess of power available over that required by the supercharger. The exhaust waste gate can be opened to by-pass part of the gases past the turbine and in that way control its output.

The reason for the use of an exhaust turbosupercharger on an aircraft engine is that, as the air pressure and hence the density drop off with altitude, the engine intake manifold pressure and density and hence power output drop off correspondingly. If an engine-driven supercharger is depended upon to do all the supercharging work, the engine brake output is reduced by the power input to the supercharger. A supercharger designed to give a pressure ratio of about 2:1 is geared to the crankshaft in all of the larger aircraft engines to make it possible for them to deliver their full rated output at low to medium altitudes. An exhaust turbosupercharger may then be added to maintain full rated engine output to high altitudes without an increase in specific fuel consumption.

Buchi Turbochargers. Diesel engines offer a much more attractive field than spark-ignition engines for supercharging with gas-turbine power, partly because the diesel exhaust temperatures are much lower and hence a turbine-bucket material is not a serious problem, and partly because excess air can be used for scavenging the clearance volume during a long valve-overlap period without appreciable loss of fuel. It also happens that the additional power required by the supercharger for the scavenging air is largely offset by the extra output from the turbine as a result of the higher gas flow rate. Since engine frictional losses increase relatively little with engine output at constant rpm, the ratio of brake to indicated output increases as the amount of supercharging is increased with this system and hence the specific fuel consumption decreases somewhat.

An exhaust turbocharger of the Buchi type is shown in Fig. 224. The turbine wheel and centrifugal supercharger impeller are directly connected on the same shaft. All air supplied to the engine flows through the supercharger, and all the exhaust gas is discharged through the turbine. The Buchi system of exhaust turbosupercharging an engine differs from that for aircraft engines in that a part of the kinetic energy imparted by the exhaust blowdown is utilized. This is accomplished by dividing the cylinders into groups and connecting each set of those whose exhaust-valve-opening periods do not overlap to one of the turbine nozzles. Thus "slugs" of exhaust gas passing out of the cylinders at high velocity are able to travel to the turbine buckets without slowing down very much so that their kinetic energy is not dissipated in eddies in a large open manifold. The turbine nozzle box serves only to control the direction of the gas flow into the wheel. Since no expansion takes place in the nozzle box, the pressure drop across it is small. The exhaust manifold required is not too complex and is readily adaptable to many engine types.

For example, in a four-cycle eight-cylinder in-line engine having the firing order 1-7-4-3-8-2-5-6, four exhaust passages would be provided, one each for cylinders 1 and 8, 7 and 2, 4 and 5, and 3 and 6. Very often all the passages are incorporated in a single manifold of cylindrical section in which partitions separate the passages for the various cylinders or groups of cylinders. It should be mentioned also that the cross section and length of the exhaust and intake passages must be carefully designed to obtain the full benefits of the arrangement. If this is done, good scavenging can be achieved over a considerable range of speed and load.

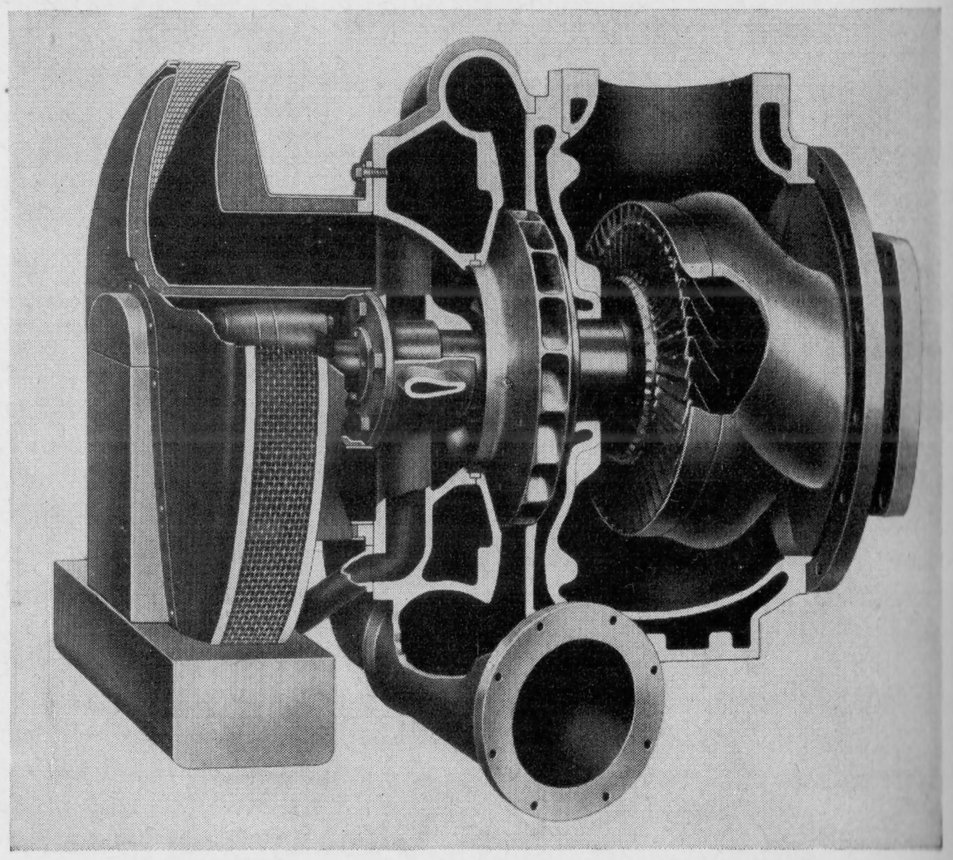


FIG. 224. Section through a Buchi-type exhaust turbocharger for a six-cylinder engine. Air enters the circular screen at the left and flows inward and to the right into the impeller, thence outward through the diffuser and the volute to the discharge flange in the center foreground. The exhaust gases from the engine enter the unit from the right through a manifold divided into three passages. After passing through the turbine blades, the gases discharge through the port above the turbine wheel. (*Elliott Company.*)

Gearing of Exhaust Gas Turbines to the Engine. Neither of the exhaust turbines described above delivers power to the engine. Their principal function is simply to increase the power capacity of the piston engine by supercharging. The possibility of coupling the turbine to the engine through a multispeed gear has been given considerable attention for large-aircraft applications, but only experimental units have been built up to the time of writing. This is an application to which the compound engine-turbine combination seems to be well suited. It happens that for aircraft use the power requirements of the propeller vary approximately as the cube of the rpm and hence match the output characteristics of the turbine.[11,12] Thus the problem of control is far more simple than it would be in the case of a truck application, for example. From the standpoint of complication, gearing the turbine to the engine represents little increase in the amount of mechanical equipment required over a conventional aircraft-engine and exhaust-turbosupercharger installation. The problems of control are more complex, however.

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Problems

1. Taking a simple open-cycle gas-turbine power plant having turbine and compressor efficiencies of 80 per cent and a maximum allowable turbine inlet temperature of 1500°F, sketch the form of curves for the following as a function of pressure ratio: (*a*) temperature rise in compressor; (*b*) temperature rise in combustion chamber; (*c*) heat released per pound of air by combustion (with constant

compressor and turbine inlet temperatures); (*d*) power required for compression per pound of air per minute; (*e*) net useful output of the cycle; (*f*) thermal efficiency of the cycle.

2. Expressing all data in terms of Btu per pound of air per minute, determine the following: (*a*) the power available from the isentropic expansion of air at 1400°F and 88.2 psia to 14.7 psia; (*b*) the power that would be delivered by a turbine wheel for the above conditions if the turbine efficiency were 85 per cent; (*c*) the power required to compress air isentropically from 60°F and 14.7 psia to 88.2 psia; (*d*) the minimum compressor adiabatic efficiency that one could have and still manage to get the turbine of (*b*) above to deliver sufficient power to drive the compressor. (Use either the air tables given in Chap. III or Table III of the Appendix for the Y factor.)

3. Determine the effect of intercooling on the thermal efficiency of the cycle and on the net work obtained per pound of air by comparative calculations for the two following cycles: (*a*) simple open cycle with a pressure ratio of 6:1, compressor inlet temperature = 60°F, turbine inlet temperature = 1500°F, and compressor and turbine efficiencies of 85 per cent; (*b*) a cycle similar to (*a*) but with three compressor stages with intercoolers between the first and second and between the second and third stages (assume that the cooling effectiveness of the intercoolers is 70 per cent and that they are supplied with coolant at 60°F).

4. Determine the effect of reheating on the thermal efficiency and net work per pound of air for the cycle of Prob. 3*b* with a three-stage turbine and reheat to 1500°F between stages.

5. Determine the effect of regeneration on the thermal efficiency and net work per pound of air for the cycle of Prob. 4 with a heat exchanger having a cooling effectiveness of 70 per cent. Tabulate the results of Probs. 3 to 5, and compare.

6. Using the Hottel charts, estimate the power available in Btu per pound per minute in the exhaust gas of a 14:1-compression-ratio diesel engine operating at standard sea-level atmospheric conditions with a fuel-air ratio of 0.0543 for the following sets of exhaust conditions: (*a*) no eddy losses from the exhaust port to the point where the exhaust gas has expanded isentropically to 14.7 psia; (*b*) constant-enthalpy exhaust-gas expansion from conditions at the end of the power stroke in the diesel cylinder to 29.4 psia, followed by an isentropic expansion to 14.7 psia (*i.e.*, the power obtainable from a single-stage impulse turbine); (*c*) same conditions as in (*b*) but with an 80 per cent turbine efficiency.

7. Sketch P-V diagrams for each part of Prob. 6, and indicate which most nearly represents conditions for the Buchi exhaust turbine.

CHAPTER XV

ENGINE INSTALLATIONS

An engine is only as good as its installation. Even the finest engine improved through many years of research and development will be considered a failure by the user if it is poorly installed. Since engines are judged in the field by the performance of the installation as a whole, most engine manufacturers either work out the installations for their own engines or at least advise as to the best methods of making any proposed installation. In either case they have a group of engineers who specialize in engine-installation problems. It requires a great deal of engineering work to install an engine with its accessory equipment so that it will give reliable, economical performance under all service conditions. At the same time, there must be no more than a tolerable amount of noise and vibration, and there must be good accessibility to all parts likely to require servicing.

Irrespective of the type of power-plant installation—passenger car, truck, aircraft, stationary, railroad, or marine—there are a basic set of components for which provision must be made. These are the engine itself; the engine mounting; the connection of the load; the cooling, induction, exhaust, and fuel systems; the controls; and the instruments and protective devices. Instructions for the details of operation and maintenance must also be prepared and possibly included in the installation in the form of a metal or plastic placard. Each of the sections in this chapter is devoted to one of these components.

ENGINE MOUNTINGS

Engine Mount. The first problem in any engine installation is that of engine mounting. Not only must the engine mount support the engine securely in position, but it is very important that the vibration accompanying engine operation be confined to the engine and not transmitted through the mount to the structure of the vehicle or building. If the latter takes place, the vibration and the attendant noise are both annoying and fatiguing to people within the structure. Thus the effectiveness of the engine mount in isolating engine vibration from the chassis and body of a motorcar or truck, for example,

is important both from the standpoint of comfort and from the standpoint of safety as well through its effect on driver fatigue.

Vibration Isolation. Perhaps the easiest type of engine-vibration-isolation problem to visualize is that in an automobile. Passenger-car engines are probably more free of vibration than any other reciprocating-engine type, but even in them the periodic torque impulses from the cylinders as they fire set up an appreciable torsional vibration. This is easily seen from Newton's third law—for every action there is an equal and opposite reaction. When a cylinder fires, a torque impulse acts on the crankshaft and is transmitted from the engine through the drive shaft. This torque impulse is accompanied by an equal and opposite torque reaction that must be transmitted from the cylinder block and crankcase to the mounting. Linear as well as torsional vibration is encountered in engine work. Any dynamic unbalance will set up a linear vibration in which the whole engine may tend to move horizontally and/or vertically. The pistons and connecting rods describe a complex motion that is not sinusoidal and may cause linear engine vibration with some cylinder and crankshaft arrangements; *e.g.*, six-cylinder in-line four-cycle engines theoretically can be almost perfectly balanced, but four-cylinder in-line engines cannot. An imperfectly matched set of pistons or rods, an unbalanced or bent crankshaft, or other similar factors will set up linear vibratory motion of any type of engine.

If the engine is rigidly bolted to a structure, the amplitude of vibratory motion will be the same for both the engine and the structure at the attaching point and the full effects of the engine vibration will be transmitted throughout the structure. In fact, the vibratory motion will probably be amplified at some points. If, on the other hand, a rubber pad or other flexible connection is inserted between the engine and the structure at the mounting points, the engine may be permitted to vibrate with a relatively high amplitude, while the frame will scarcely be disturbed. It is not easy to achieve this condition, and the detail methods by which such a mounting can be designed are quite complex.

Some idea of the approach that is employed can be gained from Fig. 225, which shows the amplitude of vibratory motion of a mass elastically connected to a vibrating body. The ratio of the amplitude of the mass to that of the vibrating body is shown as a function of the frequency with which the vibrating body is moving. At the natural, or resonant, frequency for a system without damping, the amplitude of the elastically coupled mass becomes theoretically infinite.

Some damping always prevails in an actual system so that the actual amplitudes become finite though large in this frequency range.[1] In examining Fig. 225 it becomes evident that the amplitude of the vibration of the mass coupled to the vibrating body can be reduced by a flexible connection only if the connection is made so flexible that the frequency of the disturbing vibration is much higher than the frequency at which the system is naturally inclined to vibrate. Note

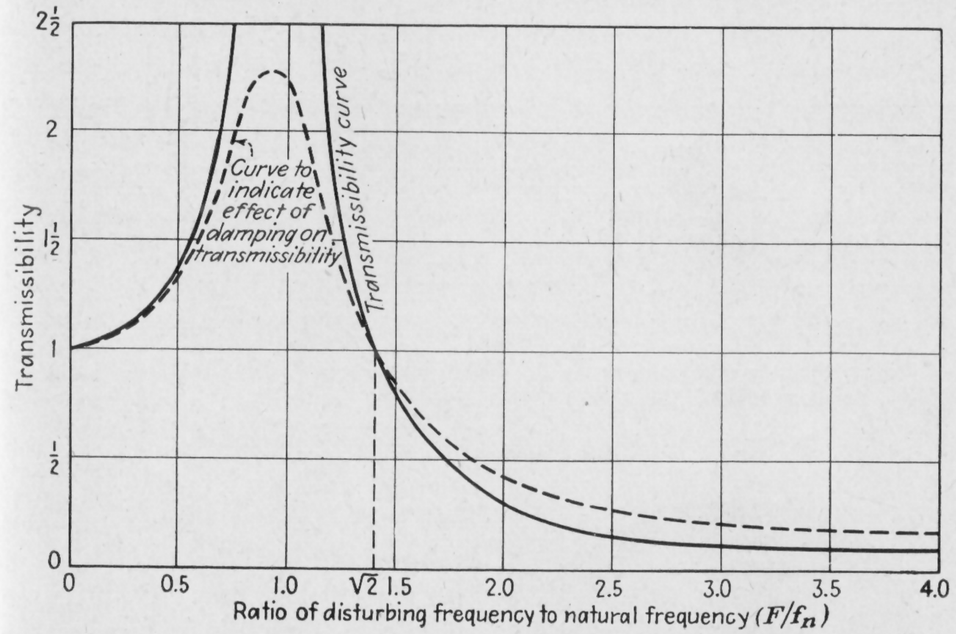


FIG. 225. Effect of the frequency of a vibrating body on the amplitude of a mass connected to it by an elastic coupling. The solid lines give values for the ideal case with zero damping while the dashed curve gives values for a representative actual case with damping. (*Lord Manufacturing Co.*)

that too much damping is undesirable because it increases the amount of motion transmitted to the elastically coupled mass in the upper frequency range and hence reduces the effectiveness of the flexible mounting.

The application of the vibration-isolation conception of Fig. 225 to engine mountings follows logically. The principal form of engine vibration is normally torsional at cylinder firing frequency.[1,2] If a flexible engine mounting is designed to permit the engine to rotate around its center of gravity, the engine itself will be displaced by the torsional firing impulse and will oscillate, absorbing the torque impulses through its inertia much as if it were a flywheel. Figure 226 shows an arrangement of flexible rubber mounting pads used for vibration isolation in a motor vehicle.[3] The flexible element in the mounting

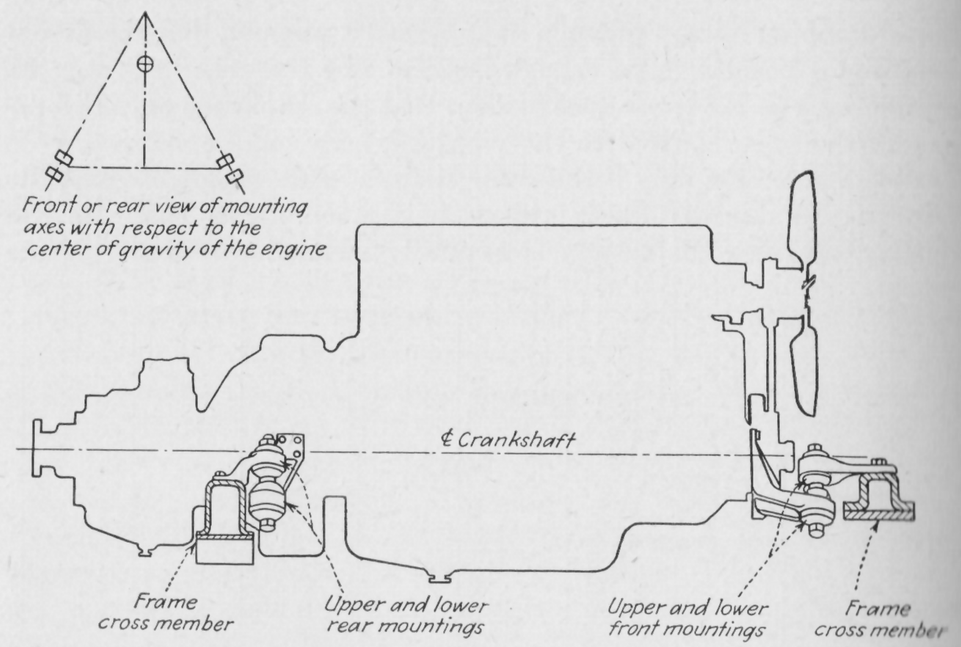


FIG. 226. Flexible mounting for an automotive engine to provide vibration isolation. (*Roche, Trans. SAE, Vol.* 50, 1942.)

is usually a rubber bushing stressed in shear. (Steel springs may be used in stationary engine installations.) A typical rubber bushing is shown in Fig. 227. It consists of two steel tubes, one vulcanized to the inside and one vulcanized to the outside of a thick-walled rubber cylinder. Such a bushing is mounted so that the normal direction of engine motion will be along the axis of the cylinder. One steel tube is fastened rigidly to the engine, and the other is rigidly attached to the mounting structure. Another bushing construction, the rubber-sandwich type, employs a rubber layer between two steel plates. This arrangement is required to stress the rubber in shear where the vibratory motion is in a plane at right angles to the axis of the bushing, as in Fig. 226.

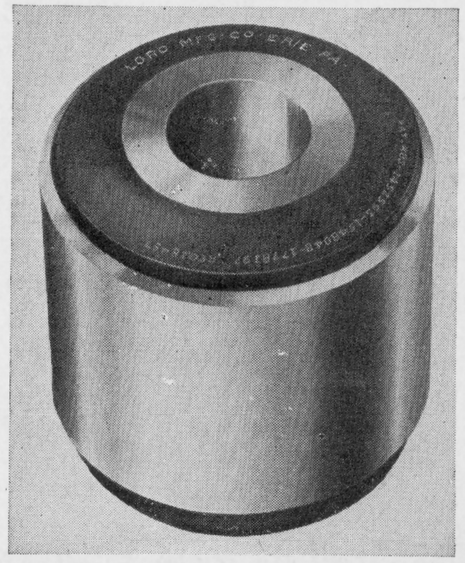


FIG. 227. Rubber bushing for vibration isolating mountings. The installation is designed so that the inner steel bushing moves axially relative to the outer steel bushing, thus stressing the rubber in shear. The bushing is much more flexible in an axial than in a radial direction. (*Lord Manufacturing Co.*)

The flexibility of the engine-mounting arrangement should be such that even in the low-speed portion of the engine-operating range the frequency of the torque variation from the firing impulses will be at least twice the natural frequency of the engine on its flexible suspension. Under very low-speed idling conditions the engine may describe large amplitude motions, but this is usually not objectionable, for the disturbing forces are small. The condition is difficult to avoid; for the natural frequency of a given system can be reduced only by increasing the flexibility of the suspension, and there are practical limits to the amount of this flexibility that can be permitted. Relative movement between the engine and the muffler of, say, 0.250 in. might be tolerated, whereas a movement of, say, 0.400 in. might cause a serious amount of service trouble.

Human Response to Vibration. The sensitivity of the human body to vibration varies from one person to another and in a given person from day to day. The source of the vibration—the floor, walls, or seat—also has its effects. Lippert[4] has correlated the results of many investigators and run some additional tests to substantiate his conclusions. He found that vertical motion of the floor is the most typical form of vibration and that human response to it could be expressed as a function of amplitude and frequency as indicated in Fig. 228. The higher the frequency, the lower the amplitude for a given degree of annoyance from the human-reaction standpoint.

Vibration Theory. The theory of vibrations is a whole subject in itself and is covered by a number of excellent texts such as that by Den Hartog.[1] Inasmuch as vibration problems not only are highly specialized but require the attention of only a small number of engineers, a further treatment of this complex subject will not be included in this text. It will suffice for most engineers if they recognize the need for and the nature of vibration-isolating mountings. They can then provide the necessary flexibility in other installation components and thus permit the engine to move with respect to the mounting structure.

ENGINE COUPLINGS AND DRIVE SHAFTS

Shaft Torsional Vibration. Closely related to the matter of engine mounting is that of the connection between the crankshaft and the load. Two main types of shaft connection and arrangement are encountered. One is that in which there is little torsional flexibility between the engine and the load. It includes aircraft installations in which the propeller is mounted directly on the engine, engine-genera-

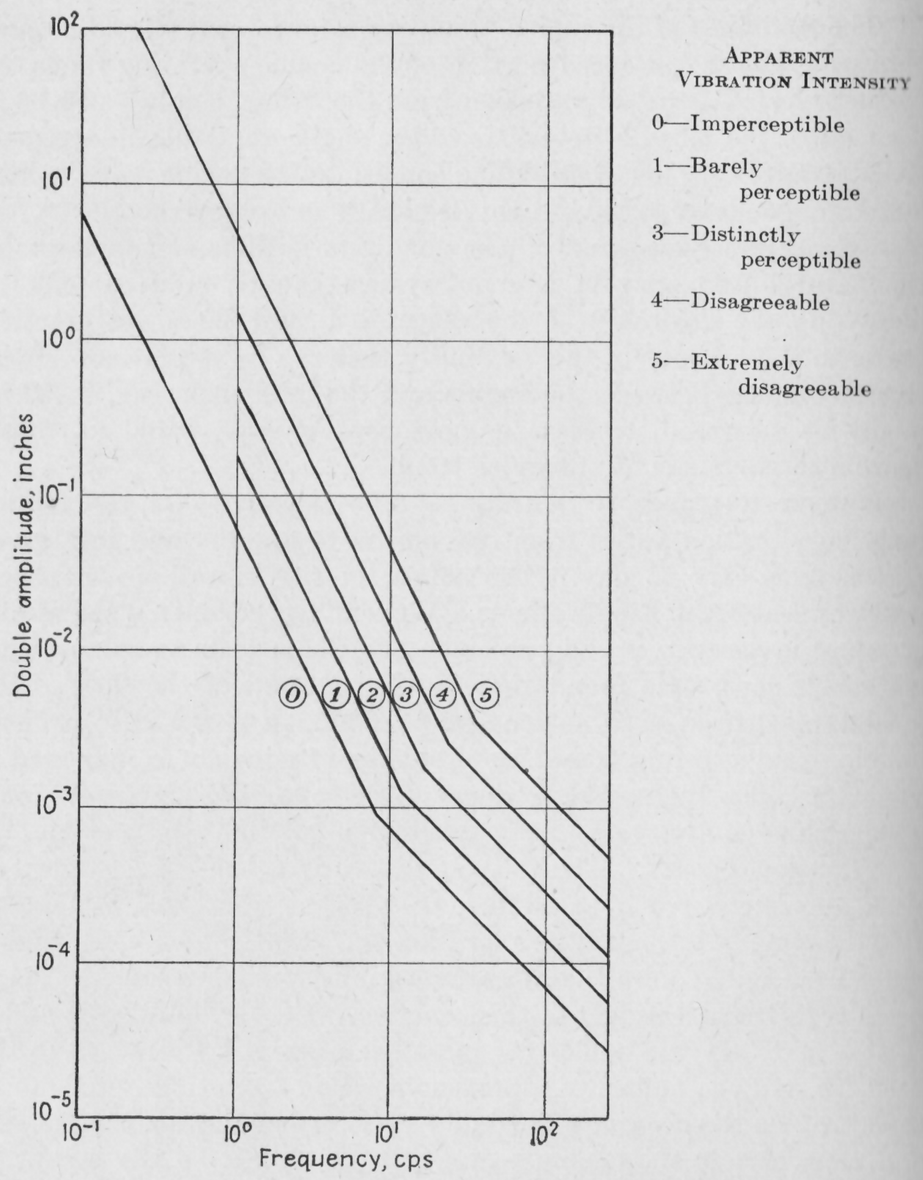


Fig. 228. Human response to vertical vibrations for the complete range of amplitudes and frequencies of major interest. (*Lippert, SAE J., Vol.* 55, No. 5, May, 1947.)

tor sets, etc. The other includes passenger-car, truck, bus, marine, and helicopter installations in which a long, torsionally flexible drive shaft connects the engine and the load. Each type has its own peculiar advantages and disadvantages, but in general it may be said that a connection of the former type may often be made without any involved vibration analysis. The latter type must always be carefully designed to provide the proper amount of torsional flexibility.

If this is done, it often actually reduces crankshaft stresses. In fact, a flexible connection may be required either to isolate engine torsional vibration from the driven element or to reduce crankshaft stresses that may arise from torsional vibration of the whole shaft system.

Provision for Misalignment. Engine couplings and drive shafts must be designed to permit some misalignment between the driving and the driven shafts. In a relatively flexible structure such as that of an automobile it is impractical to avoid a considerable amount of misalignment. Probably the simplest way to provide for this misalignment is to employ two universal joints, and this is the method generally used in automotive work. If, on the other hand, an engine is closely coupled to a machine such as a generator, it is important that a good alignment between the shafts be maintained and that a coupling having some flexibility to accommodate the unavoidable misalignment be provided. The former condition cannot be met if the engine is mounted with some flexible suspension between it and the base that supports the driven machine. Consider a portable generator set, for example. The engine and generator must be rigidly bolted to a common frame to keep them in good alignment. The complete unit, however, might be mounted on a flexible suspension to isolate vibration originating in the engine from the truck, ship, or building in which the set might be installed.

COOLING SYSTEMS

Most of the internal problems of cooling systems were covered in Chap. XI, Cooling. This section is intended merely to relate that material to the engine installation as a whole.

Cooling-air Flow. Except in marine and most stationary installations, an air stream ultimately carries off the heat transferred through the engine cylinder walls irrespective of whether the engine be air- or liquid-cooled. The motion of the vehicle may be depended upon to produce the required cooling-air flow, as in a motorcycle. If this air flow rate is not adequate, it may be augmented by employing a fan to increase the air flow rate, as in a passenger car. Again, the propeller serves as a sort of fan for the ground cooling of conventional aircraft. Slow-moving vehicles such as trucks and tanks must have a fan or blower of sufficient capacity to supply all the cooling air required, independent of vehicle motion. In fact, more than an adequate capacity may be required to allow for unusual conditions such as hill climbing with a strong wind from the rear. These problems are partly aerodynamic in nature and must be worked out by wind-tunnel or full-scale vehicle testing under the extremes of service conditions.

Ventilating-air Flow. The ventilating air passing over the surface of any type of engine plays an important role in cooling. It cools the oil in the pan in motor vehicles, contributes materially to oil cooling in any engine installation, keeps ignition equipment and accessories within allowable temperature limits and thus prevents charring of electrical insulation, and prevents the fuel lines and carburetor from becoming so hot that a serious amount of fuel boiling would take place. If the engine is in a confined space, as in a vehicle, forced ventilation is necessary. While a powerful air blast is not required for this purpose, the engine compartment should be designed so that there is a positive forced circulation of air over all parts that depend on the ventilating air flow for cooling.

Accessibility. Liquid-cooling systems require considerable maintenance. The filler cap must be readily accessible for checking of coolant level and refilling. The drain cock, hose connections, water pump, and fan belt should all be easy to get at for servicing.

INDUCTION AND EXHAUST SYSTEMS

Some of the problems of the induction and exhaust systems were covered in the chapter on manifolding. Others that are associated with the engine installation as a unit will be mentioned in this section. For example, the general layout of the various components deserves careful attention. The air intake should be located where it will induce only clean, cool, fresh air. The highest convenient location is usually best from the cleanliness standpoint. The exhaust should be discharged in such a way that it cannot reach either the engine air intake or personnel in the vehicle, ship, or building. Not only is the odor of the exhaust unpleasant, but the aldehydes in any engine exhaust will induce a severe headache. In spark-ignition engines the exhaust gas is also likely to contain carbon monoxide, the poisonous effects of which are well known. The entire exhaust system should be so arranged that there is a strong flow of ventilating air around it, partly to carry off gas leakage and partly to keep the metal cool and thus reduce high-temperature oxidation.

Air Cleaners and Silencers. Dust entrained in the air supplied to an engine will cause very high wear rates. Agricultural, earth-handling, and ordnance equipment present especially severe problems.[5] Pistons, rings, and cylinders are most affected, but the dust particles together with the products of abrasion tend to mix in with the oil and be carried all through the engine so that other parts are also exposed to the abrasive material. The rate of wear caused by

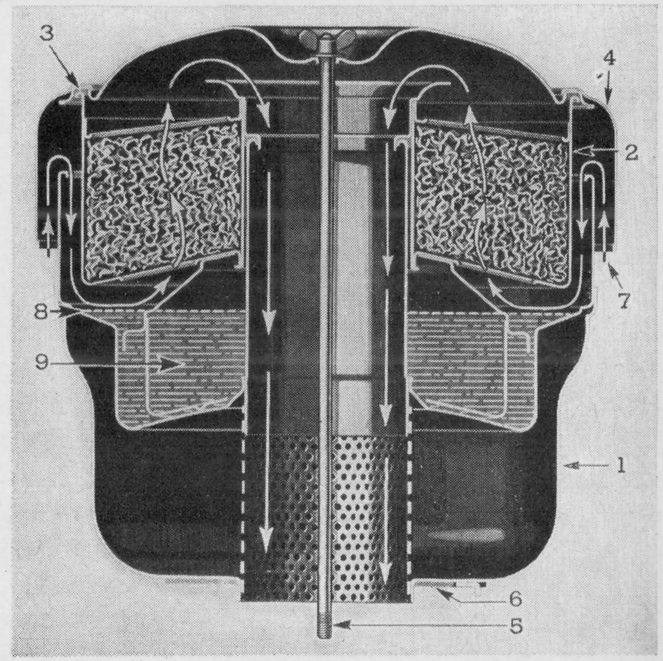
dust varies with both the size and the nature of the dust material. Particles of quartz appear to cause more damage than any other mineral common in dust. As might be expected, the rate of wear increases rapidly with dust-particle size, but even particles in the sizes as small as those in the range from 0 to 5 μ will cause excessive wear.[5] Since air filters vary in their effectiveness with the size of the dust particle, the larger particles being the more easily removed, tests on filter effectiveness must include work with the complete range of dust-particle size.

Several types of air cleaner have been used. The centrifuging action associated with the movement of air around a bend is effective in removing the larger particles. Most of the automotive air cleaners make use of what is called an *oily-maze*, or *viscous-impingement*, type of filter element. This consists of a closely packed mass of wires that have been coated with oil. The dust particles tend to follow straight-line paths in passing through a filter of this type, and hence most dust particles strike oil-covered wires before passing very far through the maze and become trapped in an oil film. When the oil films on the wires become loaded with dust, the filter element can be removed, cleaned, dipped in clean oil, and reinstalled after the excess oil has been drained off.

Under more severe dust conditions, a filter of the type just described requires too frequent servicing. This can be remedied by incorporating an oil bath in essentially the same form of air cleaner. A unit of this type is shown in Fig. 229. Air passing into the cleaner is directed downward against an oil bath at a fairly high velocity so that the larger dust particles and some of the smaller are centrifuged out and entrained in the oil. Most of the rest of the particles are then removed in the viscous-impingement filter element as the air is directed back upward. A continual washing action is also maintained over the filter element, for some oil is picked up by the air stream passing over the surface of the oil bath. Engine vibration and vehicle motion contribute to this action by causing a considerable amount of splash out of the oil bath. These entrained splash oil droplets impinge and deposit on the filter element. The accumulated oil film tends to thicken and then drain down the wires to the walls, from which it flows back into the oil bath. Continual circulation of oil in this way serves both to wash the filter element and to keep the wires covered with an oil film. This type of filter has been found to be very effective and is widely used on all types of engine.

The high velocities in the induction system tend to produce a vari-

ety of objectionable noises. These include a hissing in the carburetor at part throttle and a "roar" from an organ-pipe type of resonance in the cylinders under conditions near full-throttle operation. It has been found that these noises can be removed quite effectively by designing the air cleaner in the form of two resonant chambers placed in series, with their volumes so proportioned that the noises are absorbed. These chambers may be seen in the air cleaner of Fig. 229.



1. Air cleaner bowl
2. Filter element assembly
3. Gasket, cover to element
4. Cover
5. Bolt
6. Gasket, bowl-to-air intake
7. Air inlet
8. Oil level
9. Oil reservoir

FIG. 229. Section through an oil-bath air cleaner. (*General Motors Corp.*)

Air pulsations from the engine are damped out as they move upward past the perforations that communicate with the air chamber in the lower portion of the air-cleaner bowl. Their size and shape for maximum effectiveness must be worked out for each particular combination of cylinder and head design, intake manifold volume and shape, carburetor, and air cleaner. The mechanism of the muffling action is based on the same sort of vibration theory as that mentioned under Engine Mountings.

EXHAUST SYSTEMS

Mufflers. One of the most important functions of an exhaust system is to quiet the exhaust noise. When it is remembered that

roughly one-third of the heat energy released in the engine leaves with the exhaust and that much of that energy is expended in the exhaust blowdown, it is easy to see that there is ample energy available to set up intense sound waves. The gases ejected from the cylinder at sonic velocity by each exhaust blowdown set up a sharp pressure wave that travels out the exhaust pipe at sonic velocity. In so doing, the wave excites a violent organ-pipe type of vibration that resonates in the exhaust system between exhaust impulses. This effect is shown in Fig. 230, an oscillogram for pressure vs. time at a

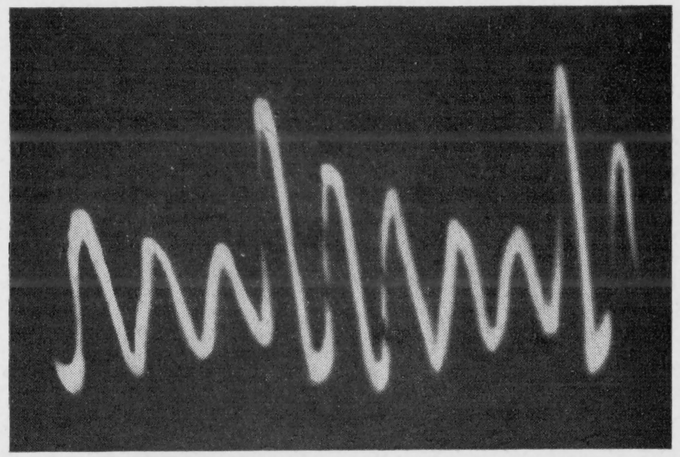
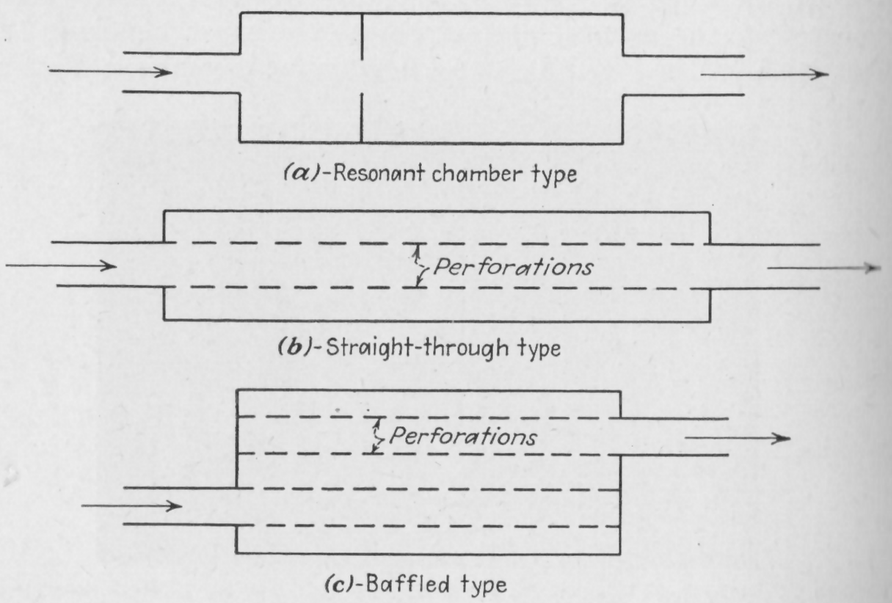


FIG. 230. Alternating pressure near the exhaust port of a one-cylinder four-cycle gasoline engine operating at 1,800 rpm under full load with an exhaust pipe 4.0 ft long and no muffler. (*Burgess-Manning Co.*)

typical point in an exhaust pipe. Each of the two high peaks represents a primary exhaust pressure wave. The four pressure fluctuations decreasing in amplitude between the two peaks show a damping of the gas-column motion induced by the exhaust pulse indicated by the first peak.

Quieting of the exhaust noise may be accomplished in several ways, one of the most effective being the use of resonant chambers designed in much the same manner as those in the carburetor air cleaners and silencers. One such arrangement is shown in Fig. 231*a*. Another type of muffler is the straight-through type shown in Fig. 231*b*. Perforations in the exhaust pipe where it passes through the muffler allow a pressure wave traveling down the pipe to spread radially outward into a chamber. The energy in the wave is reduced progressively in this way as it passes through the muffler. A third type of muffler is shown in Fig. 231*c*. It combines the characteristics of both types in that a pressure wave is broken up both by diffusion through the holes

in the inlet and exhaust pipes and by the effect of a large volume chamber between the two pipes. A still more elaborate form is shown in Fig. 231*d*, in which a series of chambers are connected by perforated pipes. An arrangement of this type, while complex, may be required in some stationary installations to meet exacting noise-level requirements.



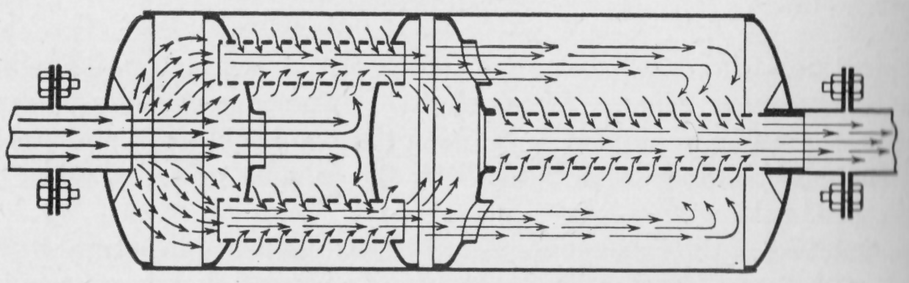


FIG. 231. Types of muffler for engine-exhaust systems.

The effect of a muffler similar to that in Fig. 231*d* is indicated by the oscillogram in Fig. 232. It was taken with a muffler on the same engine as that for which Fig. 230 was obtained. The diagram is to the same scale as Fig. 230. It again shows two exhaust impulses, but the pressure variation is much smaller. The effectiveness of the muffler in decreasing both the amplitude of the primary exhaust pressure wave and the rate of pressure rise in the wave front is clearly

evident from a comparison of the two. Not only is the amplitude of the pressure variation greatly reduced, but the dominant frequency has been dropped to a value close to the lower limit of the sonic range.

The diagrams in Fig. 230 and Fig. 232 show the pressures at the engine exhaust port. Similar diagrams showing the pressures in the first two resonant chambers of the muffler for the case with the muffler installed are shown in Fig. 233. The pressure variation at the exhaust-pipe outlet would be even smaller and the noise level correspondingly low. In comparing these four diagrams it is evident that a well-designed muffler will reduce not only the pressure variation at the exhaust-pipe outlet but that at the exhaust port as well. Thus, if a long exhaust pipe is required for a multicylinder engine in which exhaust events overlap, it may be possible to reduce the effective exhaust back pressure as each exhaust valve closes by using a well-designed muffler.

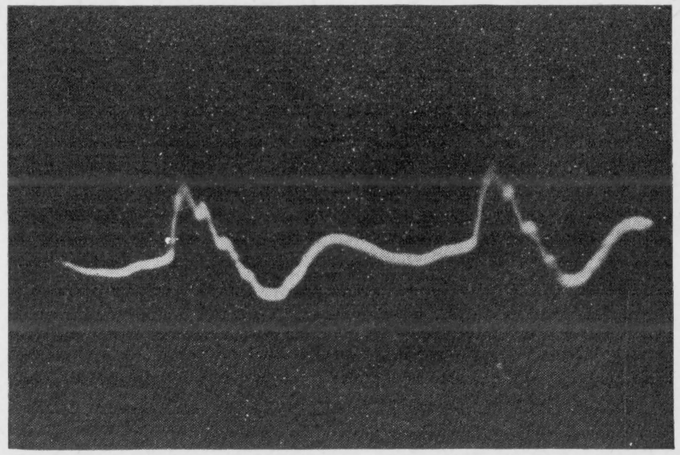
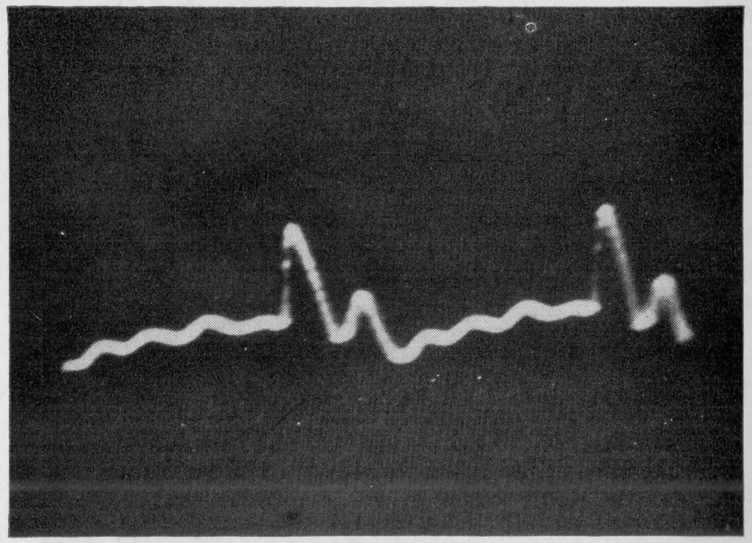
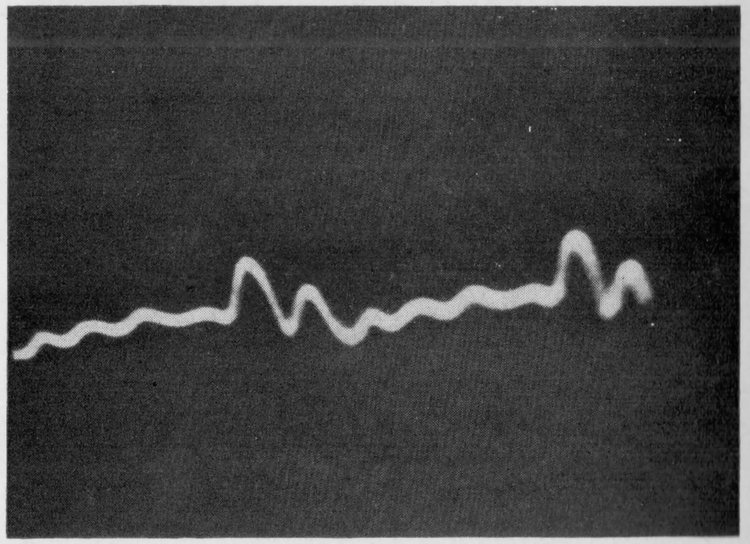


FIG. 232. Alternating pressure near the exhaust port of a one-cylinder, four-cycle gasoline engine operating at 1,800 rpm under full load with a muffler similar to that of Fig. 231*d* at the end of an 11-in.-long exhaust pipe (no tailpipe). (*Burgess-Manning Co.*)

Water Cooling of Exhaust Systems. In engine-dynamometer-test work as well as in stationary and marine installations it is usually necessary to employ a lengthy exhaust system. Partly because of the thermal-expansion problem and partly from the standpoint of the safety and comfort of personnel who must work around the engine it is very desirable to keep this exhaust pipe cool. One method is to water-jacket the exhaust manifold. Another simple and more effective procedure is to inject water, which, by evaporating, will absorb enough heat to cool the exhaust. Since an excess of water is usually injected, a low point should be provided in the system, with an ample



(*a*)



(*b*)

FIG. 233. Pressures in the first two chambers in the muffler of Fig. 231*d* while operating in the exhaust system of the engine of Fig. 232. (*a*) First chamber, (*b*) second chamber. (*Burgess-Manning Co.*)

drain through a trap. The drain will prevent water from getting into the engine cylinders if the water sprays into the exhaust manifold are accidentally left on when the engine is stopped.

Other benefits also accrue from cooling the exhaust. The reduction in temperature serves to reduce the volume flow and hence reduces both the exhaust noise and the size of exhaust pipe required. The noise reduction is particularly noticeable.

Engine-breather Fumes. While not strictly a part of the exhaust system, the engine breather serves to vent fumes from the crankcase. Ventilation should be provided around the breather outlet to carry the fumes away from personnel in the vicinity of the engine, for such fumes consist largely of cooled exhaust gases. Care should also be taken to place the breather in a region as dust-free as possible, to prevent dirt from entering the crankcase.

FUEL SYSTEMS

The principal parts of a representative fuel system are shown in Fig. 234. A fuel pump is usually required to ensure an adequate

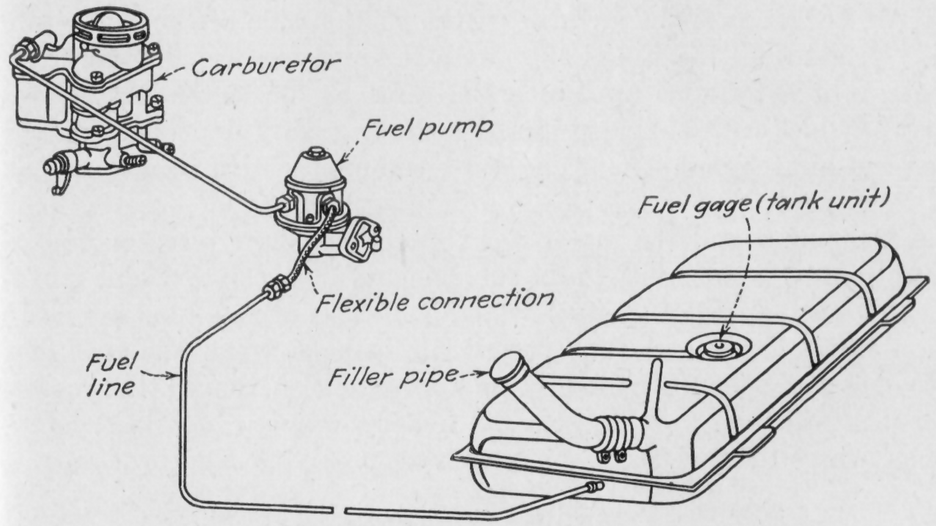


FIG. 234. Fuel system for a passenger car. (*Ford Motor Co.*)

supply of fuel at the engine, for it is desirable to place the tank at a point remote from engine radiant heat to minimize both the fire hazard and evaporation losses. A strainer and water trap are commonly placed in the fuel line immediately ahead of the fuel-metering device to prevent foreign material from entering that sensitive element and interfering with its metering.

Vapor Lock. A major consideration in the design of any gasoline-supply system is vapor lock.[6] Since the initial boiling point of gasoline is of the order of 100°F, it is not hard to see the possibility of large gas bubbles forming in the fuel line where it is close to hot parts. These bubbles may collect in the fuel pump and cause vapor lock. Since the volume of the vapor is roughly 150 times the volume of the liquid gasoline, the capacity of the pump becomes hopelessly inade-

quate if it must handle an appreciable amount of vapor and engine stoppage may result.

The installation may be designed to minimize the possibility of vapor lock in three ways.

1. Keep fuel lines in cool, well-ventilated regions and, above all, away from the exhaust system.

2. Place the fuel tank on a level with or only slightly below the fuel pump.

3. Provide a line of more than adequate size with a minimum number of bends and fittings in order to keep the pressure drop between the tank and the fuel pump to a negligible value even at the highest fuel flow rates.

Air Lock. A slight air leak on the suction side of the fuel pump may produce air locking of the pump in any type of engine-fuel system. The symptoms are much the same as for vapor lock except that troubles with engine stoppage are not greatly dependent on the ambient-air temperature. The most common source of an air leak of this sort in motor vehicles is a deteriorated rubber hose between the fuel pump and the metal fuel line. This short piece of hose is used to provide flexibility in the fuel line and thus isolate engine movement from the rest of the fuel system. Cracks or small holes tend to form in the hose and thus permit air leakage when the fuel flow becomes sufficiently high to give an appreciable suction in that region. Air lock will tend to be most acute under low-speed high-load conditions, where the fuel pump operates most nearly at capacity.

ENGINE CONTROLS

If the engine is mounted on a flexible suspension to isolate vibration, the control linkage must be arranged in such a way that engine movement will not affect the control settings. This may be accomplished by bringing the control rods in to the engine at a point close to the center about which it oscillates (engine motion is principally torsional about the crankshaft in most engines). Another method practically as satisfactory is to use a long link perpendicular to the plane of engine motion (*i.e.*, parallel to the crankshaft) between the control lever on the engine and the last control-linkage support point on the mounting structure. Yet another method is to employ flexible control cables of the push-pull type and attach the engine end of the cable sheath rigidly to the engine. The inner sliding wire or cable can then be pushed or pulled relative to the outer sheath to produce the desired control motion.

INSTRUMENTS

The most important instrument for any engine installation is an oil-pressure gauge, for nothing will damage an engine so quickly or so severely as operation without oil. An indication of coolant temperature in liquid-cooled engines or of cylinder-head temperature in air-cooled engines is probably the next most important from the standpoint of engine protection. If a battery-coil ignition system is used, an indication of the battery charging rate or the condition of the battery charge is important to give a warning of an impending ignition failure caused by a run-down battery. Other instruments such as a tachometer, a gauge to indicate the contents of the fuel tank, an oil-temperature gauge, and exhaust-temperature thermocouples may also be provided depending upon the nature of the installation and the instrumentation expense that is justified.

ACCESSIBILITY

Engines require a considerable amount of maintenance work, the cost of which is greatly dependent upon the accessibility of the parts involved. Since service troubles are difficult to visualize when the engine is in the design stage, all parts that might conceivably require servicing should be made accessible. A good example of a minor servicing job that was made into a major operation by poor accessibility is that of a portable diesel generator set made by a leading engine manufacturer for use by the armed services in the Second World War. For some reason a shear pin in the oil-pump drive shaft was a little too light and tended to fail about every six months. While at first glance it would appear that the oil pump could be removed without too much trouble, there was actually an interference with the generator housing that prevented removal. The only way to handle the job was to remove the whole generator weighing several tons—to service an oil pump weighing a few pounds! Needless to say, the necessary hoists for handling the generator were usually improvised only with difficulty.

In general, it may be stated that every accessory and every item of engine equipment should be readily accessible for removal and replacement in any type of installation. All elements requiring adjustment such as the oil-pressure relief valve, tappets, and thermostat should of course be easy to get at for servicing. In addition, the oil pan, cylinder head, main and connecting-rod bearings, and valves should be accessible for removal and inspection. A list of various items the accessibility of which should be considered is given in

Table 19. The only way to make certain that so many different items can be reached in a compact installation is to make a mock-up while the installation is in the design stage and use wooden parts where necessary. If this were done for every installation, the engineering profession would enjoy a better reputation among mechanics.

TABLE 19. ACCESSORIES, EQUIPMENT, PARTS, AND ADJUSTMENTS FOR WHICH GOOD ACCESSIBILITY SHOULD BE PROVIDED IN THE INSTALLATION

Accessories and equipment	Servicing and adjustment	Part inspection and replacement
Carburetor	Radiator filler cap	Radiator-hose connections
Ignition harness	Radiator drain cock	Fuel lines
Distributor	Oil level gauge (dip stick)	Cylinder head
Breaker points	Oil filler cap	Valves
Coil	Crankcase oil-drain plug	Exhaust manifold
Fuel pump	Spark plugs	Fan belt
Generator	Air cleaner	Oil pan
Starter	Oil filter	Connecting-rod bearings
Water pump	Fuel strainer and water trap	Main bearings
Thermostat	Tappets	Timing gear (or chain)
Injection nozzles	Oil-pressure relief valve	
	Throttle and other controls	
	Timing marks on flywheel	

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CHAPTER XVI

OVERHAUL AND MAINTENANCE

One measure of the importance of engine-overhaul and -maintenance work is given by its cost. For a passenger car the total cost during the life of an average engine is usually about equivalent to the original cost of the engine. In the truck, aircraft, railroad-, stationary-, and marine-engine fields the much more intensive usage and the lower obsolescence rate make maintenance and overhaul costs a far greater item. Much can be done to keep these costs to a minimum both by proper engine-installation design and by proper engine operation and servicing procedures. This chapter is intended to present the more important items requiring attention, together with the factors causing them, their symptoms, and the usual service procedures employed for their correction.

Engines may require attention either to prevent trouble from developing or to correct it after it has occurred. While troubles with the ignition system and carburetor or fuel-injection systems are prominent in engine maintenance work, they are not covered in this chapter. They have been omitted partly because they were discussed to some extent in Chaps. VI and VII and partly because each component made by any given manufacturer is inclined to have varieties of trouble peculiar to itself. Various makes and models of engine, on the other hand, present much the same types of service problems. Deterioration in engine performance as a result of factors not in the fuel-metering or ignition systems may arise from wear, corrosion, engine deposits, or mechanical failure.[1] The first and second of these are the most obvious, the third is responsible for a great deal of maintenance work, while the last is an infrequent though compelling cause for engine overhaul.

ENGINE OVERHAUL

It has usually been found very much worth while to set up a carefully organized overhaul shop wherever a group of about 20 or more similar engines are operated in the same type of service. Truck, bus, and aircraft fleets are examples of this sort. The length of time for which an engine can be operated before it should be given a complete

overhaul can be looked on as a statistical problem. For example, if 1 bus engine in a fleet of 100 gives a certain type of trouble at the end of 60,000 miles of operation and 10 more cause trouble before they have been run for 70,000 miles, most of the rest will probably suffer the same difficulty before they have been run for 80,000 miles. While this particular trouble might be corrected without completely overhauling the engines, deterioration in other engine conditions will also have developed to about the same degree.

The almost simultaneous appearance of a variety of engine ailments is no accident—it is the natural result of the engine manufacturer's long program of engine development. When the first units of a new engine model are built, some types of trouble may appear after only a few thousand miles of operation, others after, say, 10,000 miles, etc. The engine manufacturer concentrates his attention on these weak parts until they have been improved to the point where they last as long as most of the rest of the engine. Thus, since it is not possible to build a high-performance engine with an infinite service life, the ideal model should resemble the famous "one hoss shay" in that each part should be just as strong and wear-resistant as all the others. Unlike the deacon, however, modern fleet operators remove their equipment from service just *before* the troubles develop instead of running it until it falls apart. Such a procedure is the most economical not only from the standpoint of direct overhaul cost but even more from the standpoint of the indirect but heavy cost of service interruptions and schedule delays.

Record Systems. One of the essential elements of a good overhaul setup is a good record system.[2] The usual procedure is to have a card made out for each new engine as it is put into service. Entries are made regularly on these cards to show the number of miles (or hours) for which it has been operated. Records are kept on the same or similar cards of regularly scheduled servicing work such as spark-plug cleaning and oil changes, as well as any troubles requiring attention. Similar records are also usually kept for such accessories as starters, generators, fuel pumps, and carburetors, and an adequate supply of spare units of these items of equipment is kept on hand. Not only do these record cards give valuable data for the maintenance of individual engines, but they are invaluable in determining the optimum intervals for overhaul. Similar record cards and entries covering part replacement at each overhaul help to define the character of engine deterioration so that steps can be taken to improve the engine. They also provide a clear basis for ordering spare parts and establishing the supply that should be kept on hand.

Steps in Engine Overhaul. The first step in overhauling an engine is the "teardown." The engine is completely disassembled, and the parts are washed with a naphtha solvent. They are then laid out on tables or racks designed for the purpose and inspected. The inspector must be an alert and thoroughly experienced mechanic, for it is he who decides what parts should be discarded, what parts require servicing, and what sort of servicing should be carried out. He might, for example, decide that the cylinder bores were satisfactory for another overhaul period, or that they should be honed, or that they should be rebored. After inspection the necessary detail-servicing work is carried out, the new parts are ordered from stock, and the engine is reassembled.

The nature of the undesirable conditions for which the inspector must watch and the servicing procedure to be followed for each type of part are interrelated. These are the subjects of the following sections. While no written or pictorial description of a condition such as "scuffing" of piston rings can serve as a substitute for experience in actual inspection work, it can help to give some notion of the problems.

FATIGUE FAILURES

One of the first precepts of machine design is that a sharp notch causes a high stress concentration in a loaded part. Failure of such a part in an engine or other type of machine is usually not the result of the application of a single excessive load as it is in a tensile test machine. Instead, after a great deal of service, a small crack starts in the overstressed region. Since the stress is accentuated at the root of the crack, it grows progressively through each load cycle imposed by the machine.[3] The number of cycles required to produce complete failure is usually many millions or billions. However, for a part loaded once each revolution, such as a tooth on the timing-chain drive sprocket, only 6 hr of operation at 3,000 rpm gives more than a million load cycles. Even in the laboratory with a carefully controlled set of loading conditions, the number of load cycles to failure is a statistical matter. That is, slight random variations in the structure of the material in the vicinity of the stress concentration cause apparently identical parts to differ somewhat in their ability to withstand repeated stress. The magnitude of this variation from one piece to another can be seen in Fig. 235, which shows the number of cycles to failure as a function of stress for a series of shafts rotated under bending load. There is a much greater difference in the life of parts in regular service. Although normal loads are not high enough to bring

about a fatigue failure in most engine parts no matter how long the service, occasional overspeed operation, shock loads from abrupt acceleration or deceleration, knock, etc., impose higher than normal stresses and hence may eventually initiate a crack. Once a crack is started, the stress concentration around it is so great that the normal load cycle causes it to progress toward complete failure.

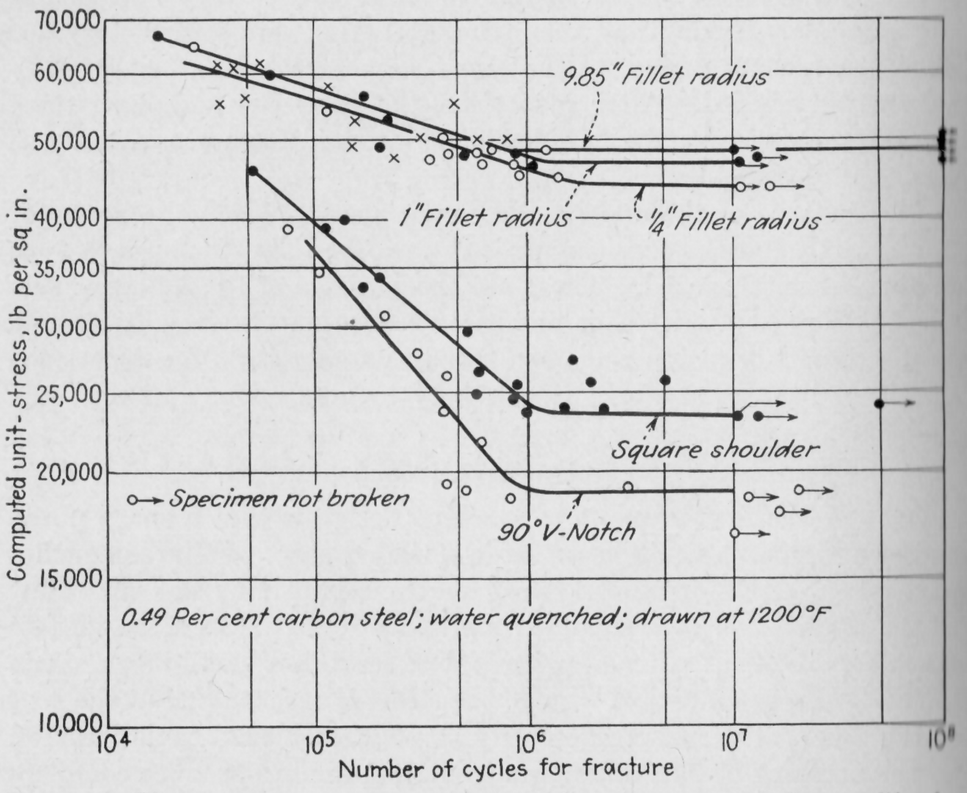

FIG. 235. Effect of stress concentration and nominal stress on the endurance life of a series of 0.400-in.-diameter bars under a bending load. The diameter of each bar was reduced to 0.275 in. by a groove. The unit stress was simply the nominal stress calculated from the cross-sectional area at the root of the groove. (*Almen, Trans. SAE, Vol.* 50, 1942.)

The appearance of a fatigue crack in an engine part is shown in Fig. 236. Note the differences in surface structure in the fractured face. These differences make it easy to identify the *nucleus*, or point of origin, of the fatigue crack. Where the crack progressed slowly, *i.e.*, near the point of origin, the apparent grain size in the fractured surface is much finer than in areas in which the crack spread rapidly. In addition, the slight relative movement of the two faces of the fracture gradually polished the fractured surface, the time available for, and hence the amount of polishing, being greatest in the vicinity of

the nucleus. The nucleus is thus made the center of a group of more or less concentric elliptical markings. Striations caused by small irregularities in the course of the crack also usually radiate from the nucleus.

The nucleus of a fatigue crack may not always be so easy to recognize as in Fig. 236. The more rapid the rate at which the fatigue

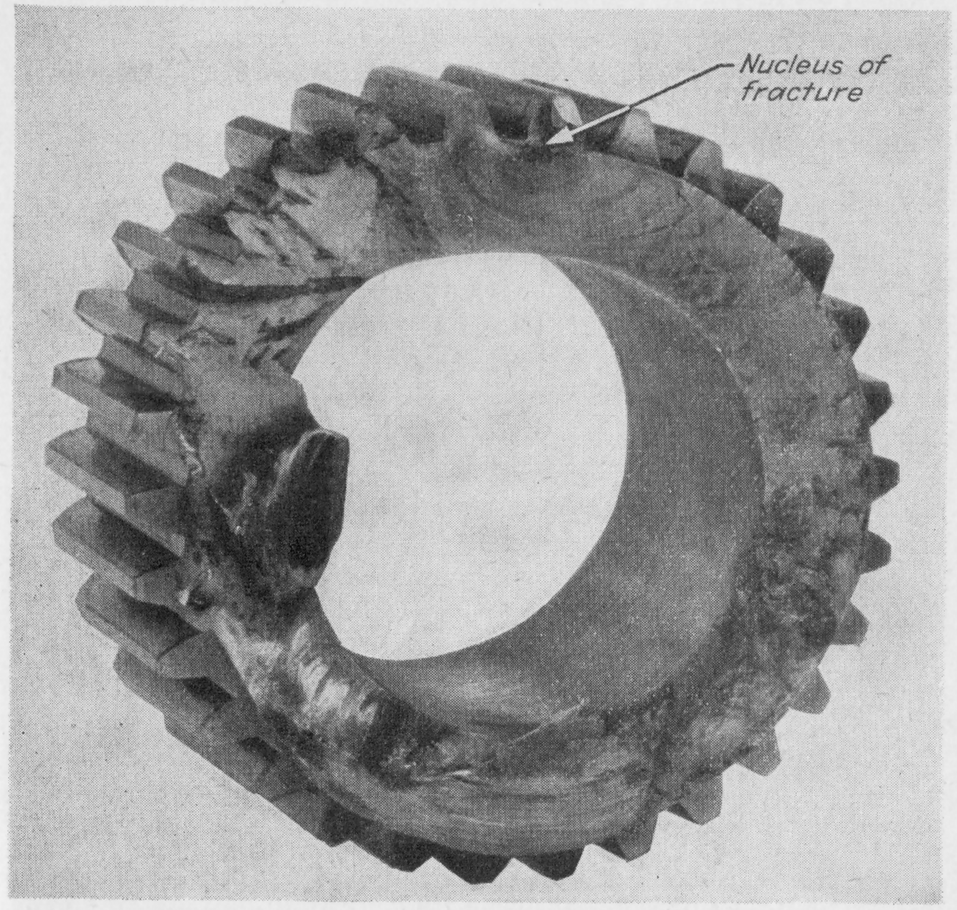

FIG. 236. Fractured surface of a part that failed in fatigue.

crack progresses, the less the difference in surface structure between the area around the nucleus and the last portions of the surface to be reached by the crack. Hardened steel parts are very resistant to fatigue and usually give a fairly well-defined *fatigue oyster shell*, as the polished area around the nucleus in Fig. 236 is called. Fatigue failures in aluminum give only a suggestion of the same appearance. Again, the denser and more homogeneous structure of forged parts gives a clearer indication of the nucleus than the somewhat porous

structure of cast parts. One characteristic of all fatigue fractures, however, is that there is no reduction in cross-sectional area where the fracture took place by fatigue. Thus, if a part cracks a portion of the way through by fatigue and then is broken the rest of the way by a single severe overload, a reduction in cross-sectional area will be evident only in the portion fractured by the single load cycle.

When an engine is overhauled, all parts should be inspected for signs of fatigue cracks so that the cracked parts may be replaced. The inspection is simplified by the fact that in virtually all types of part the crack begins at the surface in a region stressed in tension.[4] A number of methods are used to aid in finding these cracks. Steel parts may be magnafluxed, *i.e.*, strongly magnetized and washed with a light oil carrying magnetic powder in suspension. The strong magnetic field at the edges of a crack causes the powder to accumulate there and clearly indicates the presence of the crack. Aluminum parts are sometimes etched with a dilute caustic solution, the more rapid attack at the edges of a crack making it much easier to find. If etching is undesirable, the parts may be soaked in hot oil, quickly washed and dried, and then sprayed with chalk dust. Oil oozing from a crack will darken the chalk dust and indicate the crack.

Fatigue cracks are sometimes caused by inclusions, porosity, cold shuts, or other metallurgical defects in the material. More often they are caused by faulty design or manufacture, *e.g.*, inadequate fillets, toolmarks, or other stress concentrations. Improper assembly is occasionally the source of trouble. Parts may be nicked or bent by careless handling, or nuts may not be tightened properly. Uneven and/or overtightening of a set of nuts may distort a part such as a cylinder head or overload certain bolts or studs. Undertightening of a screw fastening reduces its life by increasing the range of stress variation to which the part is subjected. Consider a connecting-rod bolt, for example. If the bolt is tightened so that the tensile stress induced by tightening the nut is greater than the maximum imposed by piston and rod inertia at high speeds, the stress in the bolt will not change appreciably. If the stress set up is lower than that imposed by service loads, the stress will vary from the stress set up to the service value. Since even a high stress will not cause a fatigue failure unless it varies, it is not surprising that the service life falls off rapidly if the tension set up falls much below the maximum service stress. This effect is shown in Fig. 237, which gives a strong argument for the use of torque wrenches in engine-assembly work. Unfortunately, the boundary-lubrication conditions on the threads are variable so that

the tension set up in a bolt by tightening the nut to a given torque may vary by ± 25 per cent or more. If a particular tension must be induced in the bolt, it can be tightened to produce a certain "stretch," which in turn can be determined by measuring the bolt length with a micrometer before and after tightening.

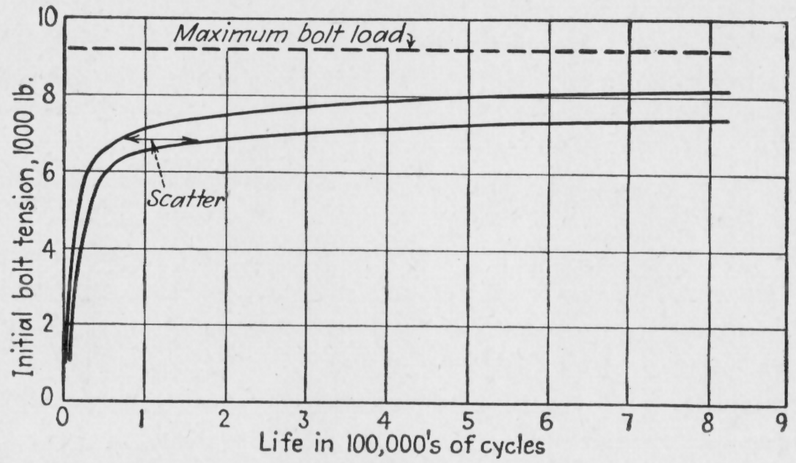

FIG. 237. Effect of the initial tension induced in connecting-rod bolts on the endurance life of the bolts. (*Almen, Trans. SAE, Vol.* 50, 1942.)

PISTON RINGS, PISTONS, AND CYLINDERS

The combustion chamber has been called the heart of the engine, for it is there that the heat energy is released and converted into mechanical work. The high temperatures under which its parts must operate makes them susceptible to a much greater range of trouble than any other engine parts. Thus it is not surprising that the combustion-chamber parts require the most attention in engine-overhaul work.

Piston-ring Wear. Piston-ring wear is the most critical factor limiting the amount of service operation between overhauls.[5–7] The ill effects accompanying it are well known—excessive oil consumption, spark-plug fouling, excessive blow-by, loss of power, and excessive fuel consumption. In view of the boundary-lubrication conditions under which the rings must operate, it is really surprising that the rate of piston-ring wear in modern engines is as low as it is. Careful development of piston-ring and cylinder-wall materials, coupled with much work on ring types and arrangements that have been designed to provide just the right amount of oil on the cylinder walls, has made possible both low wear rates and low oil consumption. Effective air filters that minimize the amount of abrasive dust reaching the rings have also helped tremendously.

Piston-ring Types and Construction. Some knowledge of piston-ring nomenclature is essential in service work. Figure 238 defines such terms as *free gap* and *side clearance*, while Fig. 239 shows the principal types of piston-ring cross section. Each type of ring has its own peculiar qualities. Compression rings are usually narrow, for a smaller seating surface gives a much better gas seal than the larger area of wide rings. Tapering the face of the ring (see Figs. 238 and 239) further reduces the width of the ring surface in contact with the

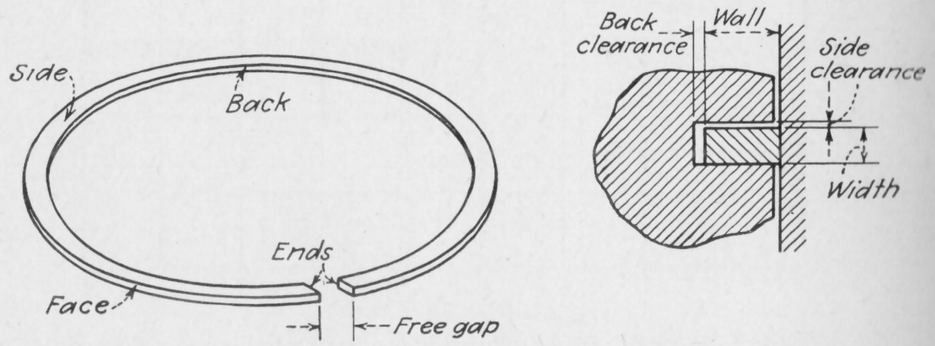

FIG. 238. Piston-ring nomenclature.

cylinder wall and thus gives an even better gas seal. It should be mentioned that, while the tension in the ring helps to keep it snugly against the cylinder wall, combustion pressures build up behind the ring to increase greatly the contact pressure between the ring and the cylinder wall. Thus gas pressure is a far more important factor than ring tension in providing a good gas seal during the high-pressure portion of the cycle. Grooved oil-control rings are usually ventilated, *i.e.*, holes or slots are provided to extend radially from the bottom of the slot through the ring to the space behind it in the ring groove. This space is also usually "ventilated" so that oil scraped off the cylinder walls may flow through radial holes in the back of the ring groove to the open interior of the piston and thence back to the crankcase.

A piston may incorporate any one of a wide variety of piston-ring arrangements, the function of any particular shape of ring varying with its location. Several typical ring arrangements are shown in Fig. 240. The top two rings are virtually always compression rings, while the lowest (and possibly the second lowest) of the rings is designed to act as an oil-scraper ring. The latter may be of a special type designed specifically for oil scraping, or it may be of a type commonly used as a compression ring but so installed as to serve as an oil

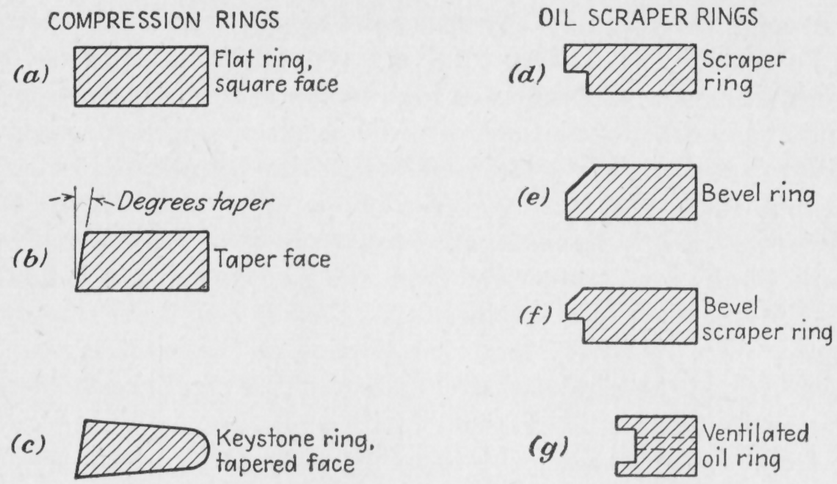

FIG. 239. Types of piston ring.

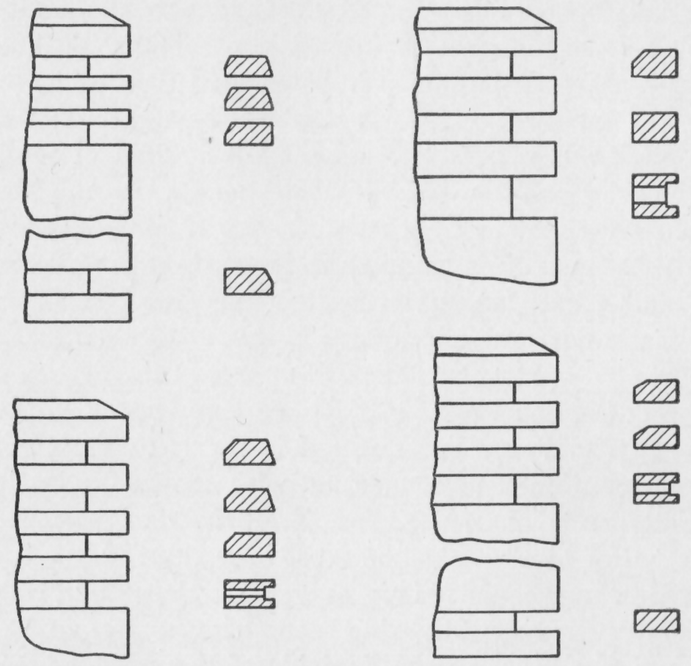

FIG. 240. Four representative piston-ring arrangements. The portions of the piston between the ring grooves are the *ring lands*. (Smith, *Trans. SAE, Vol.* 53, 1945.)

scraper. If the engine is designed to operate with high peak cylinder pressures, the number of compression rings must be increased, there usually being one compression ring for every 200 to 300 psi peak pressure at full load.

Piston rings are usually made of fine-grained cast iron. The free graphite in the cast iron (and its beneficial effect under boundary-lubrication conditions) is one of the reasons why this material has

been greatly preferred to any other. The quality of the cast iron must be carefully controlled to give a very uniform structure, good strength, dimensional stability at high temperatures, and other special characteristics small variations in which make the difference between satisfactory and unsatisfactory production ring setups.

As might be expected, the severity of operating conditions increases rapidly from the bottom of the skirt to the crown of the piston. Since the bulk of the heat transferred from the piston to the cylinder wall passes through the rings, it follows that the top ring must run hottest and carry off a relatively larger proportion of the heat flowing into the piston from the combustion chamber. This condition holds even if some piston cooling is obtained with an oil spray. Since it is farthest from the crankcase, the top ring must also operate with the least amount of lubrication. In high-performance or other engines in which especially severe operating conditions make it difficult to obtain satisfactory service life from plain cast-iron rings, a plating of tin or of hard but porous chromium may be applied to the face of the top ring. The tin is effective because it is soft enough to act as a lubricant if the oil film breaks down momentarily. This, of course, means that the tin eventually wears off. The porous chrome seems to be effective when used in conjunction with very hard cylinder-barrel surfaces, partly because oil is trapped in the surface irregularities of the chromium and partly because the two very hard surfaces rubbing together resist wear and/or scuffing if the surface-contact loads are not too great.[8]

Piston-ring Service Problems. Quite a number of different types of piston-ring trouble are encountered in service. Ring sticking has already been mentioned in connection with the carbonization, sludging, and detergency characteristics of lubricants (page 246). Ring wear was mentioned above as the most important factor limiting the period between overhauls. This may take place uniformly around the periphery of the ring if the ring rotates in the groove, or the tendency of the piston to slap from one side of the cylinder to the other at upper and lower dead center may produce a greater amount of wear in certain areas. Normal piston-ring wear produces a high polish uniformly across the face of plain rectangular rings. Wear on a taper-face ring appears as a narrow, polished band around the larger diameter edge of the ring face. If the amount or type of lubricant supplied to the ring is not adequate, *ring scuffing* may occur in the form of an area of fine parallel scratches in the direction of rubbing (see Fig. 147). If the scuffing is moderately severe, ring *feathering*

will be found, *i.e.*, a "featheredge," or burr, will form. This is similar to the burr that forms on the lower edge of a piece of metal when held against a grinding wheel. If ring scuffing is severe, scuffing or scoring of the cylinder wall will also have taken place, scoring being the more severe condition in which the scratches are deep enough to be distinct and separate. Both the burr on a feathered ring and score marks in a cylinder can be readily detected with the edge of the fingernail.

Piston rings may tend to collapse because of high-temperature creep in the metal. This condition is easily checked by measuring the free gap both before and after the ring has been operated in the engine. If the ring collapses too much, the ring tension will not be sufficient to keep it snugly against the cylinder walls. Ring flutter in the groove, gas blow-by, and excessive ring-groove wear in the piston will result. Another possible difficulty arising from high ring temperatures is *butting*, the occurrence of which is evidenced by bright worn spots at the ends of the ring, where they have been rubbing, or butting, together. Butting can be corrected by the use of a larger ring gap for the installed condition so that, as the ring expands more than the cylinder wall because of its higher temperature, there will always be sufficient gap to permit the expansion without interference between the ends of the ring.

Ring breakage may be encountered as a result of any of several conditions. Detonation, excessive ring side clearance or ring-land deflection permitting cocking of the ring in the groove, and ring sticking are the main sources of trouble. Insufficient ring gap or a ring-land failure may also cause ring breakage. If an engine is operated very long with a broken ring, the broken pieces may wear the ring lands to the point where land breakage occurs. While oil consumption and gas blow-by go up a great deal when ring breakage occurs, failure of a ring land and with it a second ring brings about very high oil-consumption and gas blow-by rates, oil fouling of the spark plug, and eventually continuous misfiring of the cylinder.

Piston Servicing. The first step in piston servicing is cleaning, for high piston-operating temperatures and long hours of operation tend to leave heavy lacquer and sludge deposits on the piston, particularly in the ring grooves (see page 246). Special solvents and caustic solutions capable of removing these deposits are available. While some care may be required in the selection of a cleaning material and process to avoid etching the pistons, this can be done and is to be preferred to scraping. The latter not only is expensive in terms

of man-hours but also tends to remove metal from the piston. After cleaning, the piston should be inspected for signs of damage such as cracked or broken ring lands, burned areas on the crown left by detonation, cracks in the piston-pin bores, and signs of scuffing or scoring. A cracked piston is generally discarded, a burned piston may be reused if the damage is superficial, while a piston with a scuffed or scored area may be stoned smooth and reused if it is not too badly marked. The width of the ring grooves and the diameter of the skirt should be checked. If the grooves are worn more than a few thousandths of an inch, the piston is usually discarded. Wear of the skirt of slotted pistons may often be compensated for by expanding the skirt in a machine and thus reducing the diametral clearance between the worn piston and cylinder walls.

Cylinder Walls. Cylinder-wall service problems are closely related to those of piston rings. High-grade cast iron has been found to be very well suited to most cylinder-wall applications, the character of the surface being naturally good for boundary lubrication. Where the cost is less important or where engine-construction considerations indicate—as in supercharged aircraft engines—a hardened steel may be employed. Hardened-steel cylinder liners have also been used in automotive-engine models to give exceptionally long service life. These liners are referred to as being of either the *dry* or the *wet* type, depending on whether or not the engine design is such that they come in direct contact with the coolant.

While scuffing or scoring of cylinder walls may occur under very severe conditions, wear is by far the more important service problem. There is apparently some fluid-wedge effect involved in piston-ring lubrication, for most cylinder-wall wear is found where the rings change the direction of their motion at the upper- and lower-dead-center piston positions. As might be expected from load and temperature considerations, the wear is greatest at the top-center position of the top ring. Cylinder wear may differ somewhat in its distribution. The most common type of wear pattern is shown in Fig. 241*a*, while a somewhat different type (often encountered in aircraft engines) is shown in Fig. 241*b*. When the wear is in the form of narrow circumferential bands as indicated in Fig. 241*b*, the worn condition is referred to as *ring stepping*. This wear condition may excite piston-ring flutter and hence be responsible for added fhp losses.

Methods of servicing cylinders during an engine overhaul vary with the type of engine, the condition of the cylinder, and the practice of the individual overhaul shop. If the cylinders are in good

condition and the amount of wear is small, they may be reinstalled without processing. However, the varnish deposit on the cylinder wall coupled with the burnishing effect of the previous set of piston rings may leave the cylinder wall so glazed that the wear rate on a new set of rings would be practically zero. This would prevent the new rings from "wearing in" (*i.e.*, adjusting themselves) to the cylinder during the first few hours of operation. This wearing-in process is essential if the minute high spots on the face of the ring are to be worn away and a good gas and oil seal is to be obtained. The cylinder-wall surface glaze can be removed by a few strokes of a fine lap or hone or possibly just with some fine emery cloth selected for the purpose. If cylinder wear is severe—usually if more than about 0.0005 in./in. of diameter—the cylinder can be honed to give a round, straight bore of uniform diameter. If this measure is necessary, oversize pistons are required to maintain the proper diametral clearance on the pistons.

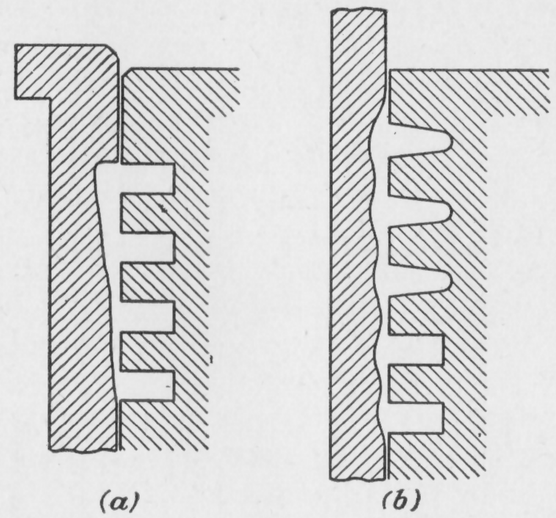

FIG. 241. Cross sections through cylinder walls showing two common types of wear pattern. The piston is shown in its top dead-center position in each case.

Cylinder Head. The only item in the cylinder head that almost always requires attention is the deposit of carbon and lead compounds on the combustion-chamber surface. This can be removed by scraping.

The core plugs in the cylinder head of a liquid-cooled engine—in common with those in the cylinder block—may require replacement, as these parts are only of sheet steel and hence are inclined to rust through.

VALVES AND VALVE GEAR

Valves and Valve Seats. While mushroom poppet valves have proved to be the most service-trouble-free type of internal-combustion-engine valve mechanism, they must operate under such severe temperature conditions that some service problems are inevitable.[9] The temperature of an exhaust-valve head, for example, may often run as high as 1200°F, while the temperature of an intake-valve head may exceed 600°F. The temperature of the valve heads would run much higher were it not for the fact that the valves are cooled by the conduction of heat from the valve heads through both the valve seat and the valve stem to the cylinder head or block.

Several types of service difficulty with exhaust-valve "burning" may arise. If the tappet clearance with the engine cold is too small, the valve stem may expand so much as the engine heats up that the valve may not close completely and gas will leak across the valve-seating surfaces. The high-velocity gas flow will cause overheating of the valve head and oxidation of the metal. Since great care is exercised to maintain the proper tappet clearances, this type of valve burning is seldom encountered. Extended engine operation at high outputs with lean mixtures may cause warpage of the cylinder head and valve seat or of the valve head and thus permit a gas leak across a portion of the seating surface. Valve burning such as that evident in Fig. 242 will, of course, follow. Corrosion of valve heads and seats by bromine compounds from the tetraethyl lead fluid is a relatively common variety of valve trouble. It is likely to be induced in an engine in which the mixture temperature is too low to give good distribution (see page 265). The usual course of this type of difficulty is general corrosion of both the valve head and the valve seat until a channel forms in one of the seating surfaces. Gas leakage and valve burning result.

Exhaust-valve heads are usually made of corrosion- and temperature-resistant steels to minimize difficulties with the various types of burning indicated above. The heads are welded to stems of less expensive steel having a higher heat conductivity to aid in keeping the valve head cool.

The twisting action of the valve springs as they are compressed or as they expand may cause spinning of either intake or exhaust valves, although intake valves are more subject to the trouble. If valve spinning occurs, the high rubbing velocity between the valve and its seat as the valve closes will induce rapid wear in the form of

circumferential grooves in the seating surfaces. If the condition is chronic, it must be corrected by modifying the valve springs, the valve-spring washer, and/or other parts of the valve gear. Valve spinning should not be confused with slow valve rotation, a desirable condition which is usually present in any valve-gear arrangement

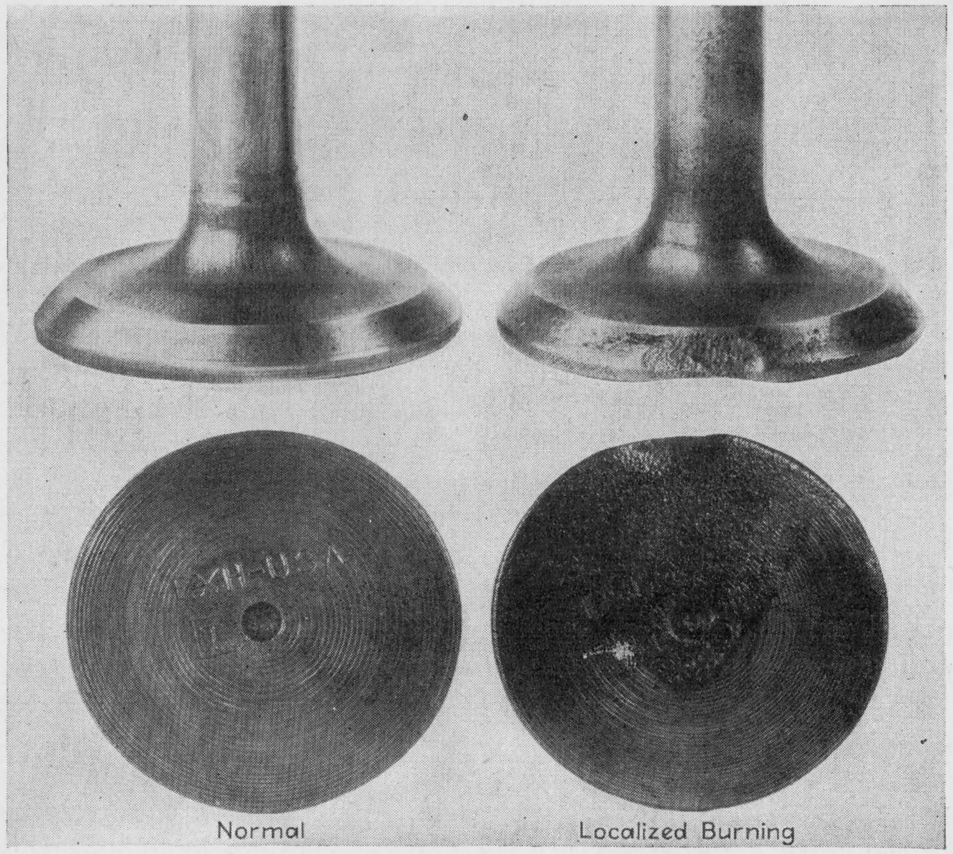

FIG. 242. Burned area in the seating surface of an exhaust valve. (*Colwell, Quarterly Trans. SAE, Vol.* 2, 1948.)

unless specifically avoided by design of the parts. Valve rotation tends to keep seat wear uniformly distributed and tends to prevent valve-head warpage. There is always a tendency toward the latter caused by a nonuniform exhaust-gas velocity distribution and hence temperature distribution around the valve during the exhaust event.

If the valve-seating velocity is too great, the seating surfaces will tend to become "pounded," or indented, as shown in Fig. 243. This condition is known as *valve-seat pounding.* Again it may occur with either exhaust or intake valves, although the latter are the more likely to give trouble. Seat pounding may be caused by improper cam

design, incorrect tappet clearances, an excessive amount of high-speed or overspeed operation, or some defect in the valve gear.

In servicing valves, they must be cleaned and the condition of each head examined. Since valve-stem wear, as well as pounding and pitting of the stem tip where it is contacted by the tappet, also presents service problems, these elements must also be inspected. If the thickness of the seating portion of the valve head has not been reduced too much by erosion or previous resurfacing and if the valve stem is not worn seriously, the valve-seating surface may be reground in a valve-grinding machine. Depending on the type of valve and the engine model, the tip of the stem may also have to be resurfaced to allow for the material removed from the valve head. The valve seats in the cylinder head may be resurfaced with a specially shaped cutter or grinding stone mounted on a shaft that pilots into the valve guide. Again depending on the engine model, the valves may be lapped into the head, although modern valve- and seat-grinding equipment usually renders this unnecessary.

FIG. 243. Intake valve with a badly pounded seat. (*Colwell, Quarterly Trans. SAE, Vol.* 2, 1948.)

Valve Guides. While it was not mentioned above, one of the most prevalent sources of valve trouble is *valve sticking* in the guides. The usual cause is a very adherent hard deposit on the portion of the valve stem that enters the outer end of the guide when the valve is in the closed position. When the valve is closing, the deposit may cause binding and sticking so that the valve spring will not close the valve. Backfiring and rough engine operation will result. If the valve-stem deposit is made up primarily of dust and lead compounds with a little oil varnish as a binder, as is usually the case on exhaust valves, removal and cleaning of the valves and cleaning of the guides are generally the only solution. A highly detergent oil may clear up valve-sticking troubles if the valve-stem deposits are gums from the fuel or tarry sludge from the oil, as is often the case with intake valves.

Valve guides are subject to considerable wear because of high valve-stem temperatures and the necessarily restricted oil supply. It should be pointed out that, if an appreciable amount of oil passes up through the clearances between the intake valves and their guides

and deposits on the valve head, it tends to carbonize there and build up a heavy deposit which interferes with the cylinder-charging process.

Valve Springs. Valve springs are likely to be subject to breakage if the engine is operated at high or overspeed conditions for appreciable periods. They may also tend to collapse or to lose their tension

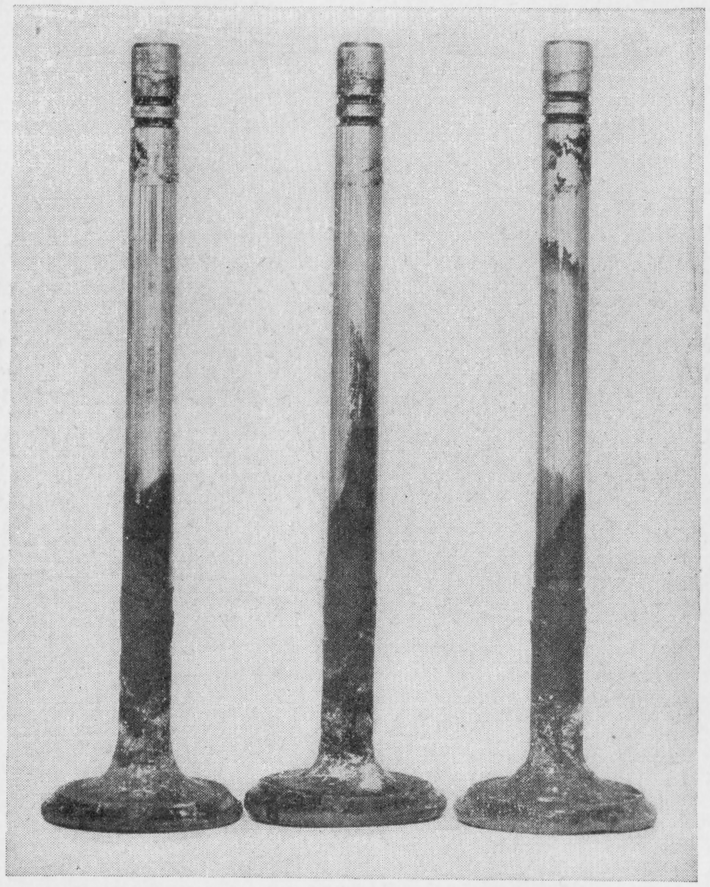

FIG. 244. Typical exhaust-valve stem deposits. (*Colwell, Quarterly Trans. SAE, Vol.* 2, 1948.)

from either overstressing or high-temperature creep. Breakage or collapsing of valve springs can be easily checked by visual inspection.

Camshaft and Tappets. The high unit bearing loads, especially for the exhaust-valve-opening event, coupled with high rubbing velocities may cause wear and scuffing of cams and cam followers. Elasticity in the valve gear is likely to cause valve "bounce," which in turn is inclined to produce high surface-contact stresses and pitting of the contact surfaces. Both wear and pitting can be easily detected by visual examination and the parts declared either satisfactory for

further use or in need of servicing. If the heat treatment originally applied to the parts gave sufficient depth of hardness, it may be possible to stone out scuff marks or worn flat spots.

SHAFTS AND BEARINGS

Connecting-rod and Main Bearings. Engine bearing surfaces are subject to wear, fatigue, and corrosion. The latter was discussed in connection with lubrication, and thus only wear and fatigue will be considered here. While wear ideally should not take place under conditions of perfect fluid-film lubrication, in practice, under what are

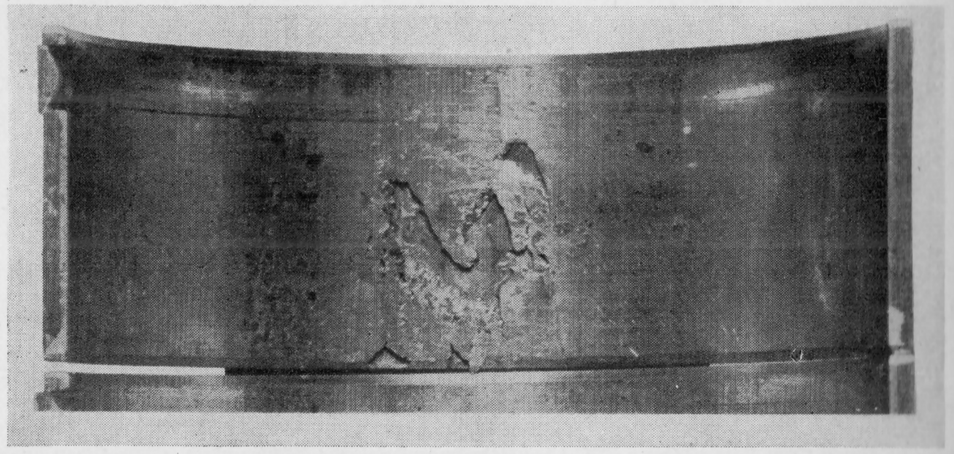

FIG. 245. A passenger-car engine connecting-rod bearing in which the surface of the lining is covered with a network of fine fatigue cracks. Portions of the lining have flaked out between the cracks in some areas.

ostensibly these conditions, it does occur. Contaminating material in the oil in the form of dust and metallic particles abraded from engine parts enters the bearings. If the particles are larger than the thinnest portion of the oil film, they tend to cut the rubbing surfaces. It is true that hard carbon particles may also enter the bearing clearance and form long, indented streaks, but these are harmless as they cause a negligible and localized loss in bearing area. While the wear rate is usually low, it eventually makes necessary replacement of the main- and connecting-rod bearings. The development of thin metal bearing inserts has made this a simple and inexpensive job.

Any given portion of a connecting-rod or main bearing is loaded intermittently because of the change in load magnitude and direction as the crankshaft rotates. This sets up a flexing action in the bearing and tends to start fatigue cracks. These cracks form perpendicular to the bearing surface and progress through the lining to the bearing

shell. After a network of cracks forms in the lining, little "islands" of bearing material tend to crack away from the shell. Figure 245 shows this condition in a pair of bearing shells from an automotive connecting-rod bearing. Pieces of lining have started to flake away from the center and lower edge of the upper half, and some of the cracks are quite pronounced. Most of the cracks in both halves are partly obscured by debris from the failed areas. Many pieces of this debris have embedded themselves in other parts of the lining after scratching and indenting the surface.

Crankshaft. Most of the wear takes place in the soft bearing linings, but some wear of the hardened crankshaft journals may take place. Journal wear is generally not uniform around the periphery; for example, crankpins tend to wear on the surface toward the inner side of the crank throw. An out-of-round condition may be corrected by any of several different methods depending on the engine type. The crankpin may be lapped down to a smaller standard diameter and new bearing inserts of an appropriate size installed, the worn area may be built up with chromium plate and reground to size, or the worn areas may be built up with weld metal and then machined and ground to size.

Accessory Drive Shafts and Bushings. Worn bushings in the engine housings can be pressed or machined out, new bushings pressed or shrunk in, and the new bushings bored or reamed to size. Worn accessory drive shafts are usually replaced, for in most engines their cost seldom justifies reclamation in the relatively few instances in which wear necessitates replacement.

SERVICING AND PREVENTIVE MAINTENANCE

Regularly scheduled servicing helps greatly to extend the time between engine overhauls and to prevent power-plant failures in service. Air cleaners, lubricating and cooling systems, filters and strainers, spark plugs and injection nozzles, etc., all require attention at regular intervals. This work should be accompanied by a careful inspection of the engine for oil leaks, loose nuts, and other indications of incipient trouble. The men operating the engines can usually detect trouble before it becomes serious if they are continually on the lookout. Their routine often calls for daily reports, part of which cover observations of such signs of trouble. When this is the case, arrangements are made so that the reported difficulties can be given prompt attention.

Much more might be said about engine servicing and maintenance,

for the field is so broad in scope. In the final analysis, the ultimate objective of most engineering work on combustion engines is high performance at low cost with the best possible reliability. The service engineer can help greatly to achieve this end by diagnosing troubles disclosed by engine overhaul, calling the attention of design engineers to these problems, setting up good servicing and overhaul procedures, and making certain that the conditions of operation make it possible to get the maximum performance and utility from the engine. Work of this kind requires an understanding of all types of engine problem from combustion and knock to fuels and lubricants, from carburetion and ignition to fatigue failures.

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APPENDIX

Table I. Useful Formulas

Thermodynamics:

Energy input to a gas during an isentropic compression without flow work:

$$\text{Work, ft-lb} = Wc_v778(T_2 - T_1) = \frac{1}{k-1} WRT_1 \left[\left(\frac{P_2}{P_1} \right)^{\frac{k-1}{k}} - 1 \right] = \frac{P_2V_2 - P_1V_1}{k-1}$$

Net work required to compress and deliver a gas:

$$\text{Work, ft-lb} = Wc_p778(T_2 - T_1) = \frac{k}{k-1} WRT_1 \left[\left(\frac{P_2}{P_1} \right)^{\frac{k-1}{k}} - 1 \right]$$

Final temperature for an isentropic compression:

$$T_2 = T_1 \left(\frac{P_2}{P_1} \right)^{\frac{k-1}{k}} = T_1 \left(\frac{V_1}{V_2} \right)^{k-1}$$

Temperature rise during an isentropic compression:

$$T_2 - T_1 = T_1 Y \qquad \text{where } Y = \left(\frac{P_2}{P_1} \right)^{\frac{k-1}{k}} - 1$$

Fluid Mechanics:

Volume flow through an orifice for conditions under which compressibility effects are negligible and where the orifice diameter is small as compared with the effective diameter of the approach passage:

$$\text{Volume, ft}^3\text{/sec} = CA\sqrt{2gH} = CA\sqrt{2g\frac{\Delta p 62.4}{12w}}$$

where C = orifice coefficient (approximately 0.97 for rounded-approach orifices and 0.60 for sharp-edged orifices)
A = orifice area, ft^2
H = pressure differential between orifice inlet and throat in terms of the height of a column of fluid having the same density as the fluid flowing through the orifice, ft
Δp = pressure differential between orifice inlet and throat, in. H_2O
w = density of the fluid flowing through the orifice, lb/ft^3

$$\text{Weight flow, lb/sec} = (\text{density})(\text{vol flow}) = CA\sqrt{335\,\Delta pw}$$

TABLE II. UNITS AND CONVERSION FACTORS

Length:

1.0 inch = 2.540 centimeters
1.0 meter = 39.37 inches
1.0 mile = 5,280 feet
1.0 mile = 1.609 kilometers
1.0 mile = 0.8684 nautical mile

Volume:

1.0 cubic foot = 1,728 cubic inches
1.0 cubic inch = 16.39 cubic centimeters
1.0 gallon = 231 cubic inches
1.0 gallon = 3.785 liters
1.0 gallon = 0.8327 imperial gallon

Weight:

1.0 kilogram = 2.205 pounds
1.0 ounce = 28.34 grams

Density:

1.0 gram per cubic centimeter = 62.46 pounds per cubic foot
1.0 pound per cubic foot = 16.02 kilograms per cubic meter
At 60°F and 14.7 psia:
Air density = 0.0765 pound per cubic foot
Density of water = 8.345 pounds per gallon = 62.4 pounds per cubic foot
Density of gasoline (approximately) = 6.0 pounds per gallon

Pressure:

1.0 atmosphere = 14.696 pounds per square inch
= 29.92 inches mercury
= 760 millimeters mercury
1.0 pound per square inch = 2.036 inches mercury
= 27.69 inches water
= 70.31 grams per square centimeter

Energy:

1.0 Btu = 777.98 foot-pounds
1.0 erg = 7.376×10^{-8} foot-pounds
1.0 horsepower-hour = 2545 Btu
= 641.7 kilogram-calories
= 0.7457 kilowatt-hour

Viscosity:

1.0 poise = 1.0 gram per centimeter-second
1.0 centipoise* = 0.01 poise
1.0 centipoise = 1.45×10^{-7} reyns
1.0 reyn = 1.0 pound-second per square inch

Kinematic Viscosity (ratio of viscosity to density):

1.0 stoke = 1.0 poise-cubic centimeter per gram
= 1.0 square centimeter per second
1.0 centistoke = 0.01 stoke
The kinematic viscosity in centistokes (ν) may be found from the value in Saybolt universal seconds (SUS) by the following empirical relation:

$$\nu = 0.22(\text{SUS}) - 0.135/(\text{SUS})$$

* The viscosity of water at 68.4°F happens to be 1.0 centipoise.

TABLE III. VALUES OF Y FOR NORMAL AIR AND PERFECT DIATOMIC GASES*

$$Y = \left(\frac{P_2}{P_1}\right)^{0.283} - 1$$

P_2/P_1	0	1	2	3	4	5	6	7	8	9
1.0	0.0000	0028	0056	0084	0112	0139	0166	0193	0220	0247
1.1	0273	0300	0326	0352	0378	0404	0429	0454	0480	0505
1.2	0530	0554	0579	0603	0628	0652	0676	0700	0724	0747
1.3	0771	0794	0817	0841	0864	0886	0909	0932	0954	0977
1.4	0999	1021	1043	1065	1087	1109	1130	1152	1173	1195
1.5	0.1216	1237	1258	1279	1300	1321	1341	1362	1382	1402
1.6	1423	1443	1463	1483	1503	1523	1542	1562	1581	1601
1.7	1620	1640	1659	1678	1697	1716	1735	1754	1773	1791
1.8	1810	1828	1847	1865	1884	1902	1920	1938	1956	1974
1.9	1992	2010	2028	2045	2063	2080	2098	2115	2133	2150
2	0.2167	2336	2500	2658	2812	2960	3105	3246	3383	3516
3	3647	3774	3898	4020	4139	4255	4369	4481	4591	4698
4	4804	4908	5010	5110	5209	5306	5401	5495	5588	5679
5	5769	5858	5945	6031	6116	6200	6283	6365	6446	6525
6	6604	6682	6759	6835	6910	6985	7058	7131	7203	7274
7	0.7345	7414	7483	7552	7620	7687	7753	7819	7884	7949
8	8013	8076	8139	8201	8263	8324	8385	8445	8505	8564
9	8623	8681	8739	8797	8854	8910	8966	9022	9077	9132
10	9187	9241	9295	9348	9401	9453	9506	9558	9609	9660
11	9711	9762	9812	9862	9912	9961	1.0010	1.0058	1.0107	1.0155

* From MARKS, L. S., "Mechanical Engineers' Handbook," 4th ed., p. 315, McGraw-Hill Book Company, Inc., 1941.

TABLE IV. SPECIFIC GRAVITIES AT 60/60°F CORRESPONDING TO DEGREES API AND WEIGHTS PER U.S. GALLON AT 60°F

°API	Specific gravity	Lb/U.S. gal	°API	Specific gravity	Lb/U.S. gal	°API	Specific gravity	Lb/U.S. gal	°API	Specific gravity	Lb/U.S. gal
0	1.0760	8.962	25	0.9042	7.529	50	0.7796	6.490	75	0.6852	5.703
1	1.0679	8.895	26	0.8984	7.481	51	0.7753	6.455	76	0.6819	5.676
2	1.0599	8.828	27	0.8927	7.434	52	0.7711	6.420	77	0.6787	5.649
3	1.0520	8.762	28	0.8871	7.387	53	0.7669	6.385	78	0.6754	5.622
4	1.0443	8.698	29	0.8816	7.341	54	0.7628	6.350	79	0.6722	5.595
5	1.0366	8.634	30	0.8762	7.296	55	0.7587	6.316	80	0.6690	5.568
6	1.0291	8.571	31	0.8708	7.251	56	0.7547	6.283	81	0.6659	5.542
7	1.0217	8.509	32	0.8654	7.206	57	0.7507	6.249	82	0.6628	5.516
8	1.0143	8.448	33	0.8602	7.163	58	0.7467	6.216	83	0.6597	5.491
9	1.0071	8.388	34	0.8550	7.119	59	0.7428	6.184	84	0.6566	5.465
10	1.0000	8.328	35	0.8498	7.076	60	0.7389	6.151	85	0.6536	5.440
11	0.9930	8.270	36	0.8448	7.034	61	0.7351	6.119	86	0.6506	5.415
12	0.9861	8.212	37	0.8398	6.993	62	0.7313	6.087	87	0.6476	5.390
13	0.9792	8.155	38	0.8348	6.951	63	0.7275	6.056	88	0.6446	5.365
14	0.9725	8.099	39	0.8299	6.910	64	0.7238	6.025	89	0.6417	5.341
15	0.9659	8.044	40	0.8251	6.870	65	0.7201	5.994	90	0.6388	5.316
16	0.9593	7.989	41	0.8203	6.830	66	0.7165	5.964	91	0.6360	5.293
17	0.9529	7.935	42	0.8155	6.790	67	0.7128	5.934	92	0.6331	5.269
18	0.9465	7.882	43	0.8109	6.752	68	0.7093	5.904	93	0.6303	5.246
19	0.9402	7.830	44	0.8063	6.713	69	0.7057	5.874	94	0.6275	5.222
20	0.9340	7.778	45	0.8017	6.675	70	0.7022	5.845	95	0.6247	5.199
21	0.9279	7.727	46	0.7972	6.637	71	0.6988	5.817	96	0.6220	5.176
22	0.9218	7.676	47	0.7927	6.600	72	0.6953	5.788	97	0.6193	5.154
23	0.9159	7.627	48	0.7883	6.563	73	0.6919	5.759	98	0.6166	5.131
24	0.9100	7.578	49	0.7839	6.526	74	0.6886	5.731	99	0.6139	5.109
									100	0.6112	5.086

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E = INTERNAL ENERGY, B. T. U., ABOVE CO_2, H_2O (VAPOR), O_2, AND AIR-N_2 AT 100°F

Q_v = INT. ENERGY OF COMBUSTION, AT 100°F., OF UNBURNED FUEL IN THE EQUILIBRIUM MIXTURE AT T. WHEN T ≤ 2880°R., Q_v = 1.5, 2.0, 3.0, AT S = 0.4, 0.6, 0.8, RESPECTIVELY

$E_s = E - Q_v$

H = ENTHALPY, $E + J(PV)$

$H_s = E_s + J(PV)$ [· $H = H_s + Q_v$]

P = PRESSURE, LBS./SQ. IN. (DASHED LINES)

V = VOLUME, CU. FT. (SOLID LINES)

S = ENTROPY, ABOVE CO_2, H_2O (VAPOR), O_2, AND AIR-N_2, EACH AT ONE ATMOSPHERE, 100°F

T = TEMPERATURE, °R = (°F + 460)

FUEL = $(CH_2)_x$

APPROXIMATE SCALE FOR HORIZONTAL INTERPOLATION BETWEEN PRESSURE OR VOLUME LINES

INTERNAL ENERGY, 100'S B.T.U.

$\frac{H_s}{100}$

$\frac{E}{100}$

$\frac{E_s}{100}$

H - INTERPOLATION SCALE

S=0.3 S=0.8

DASHED LINES, PRESSURE

SOLID LINES, VOLUME

DASHED LINES, PRESSURE

SOLID LINES, VOLUME

DASHED LINES, PRESSURE

IN THIS RANGE,	
S	Q_v
0.3	1.0
0.4	1.5
0.5	1.5
0.6	2.0
0.7	2.5
0.8	3.0

F = 0.9
PRODUCTS OF COMBUSTION OF 1 LB. OF AIR WITH 90% OF THEORETICAL FUEL, IN CHEMICAL EQUILIBRIUM 0.06098 LBS. FUEL/LB. AIR

ALL VALUES OF EXTENSIVE VARIABLES TAKEN FROM THIS CHART ARE ON A MASS BASIS AND THEREFORE ARE REFERRED TO IN THE TEXT BY LOWER-CASE SYMBOLS.

ENTROPY, B.T.U. PER DEGREE R.

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M.I.T., CAMBRIDGE MASS.

E = INTERNAL ENERGY, B. T. U., ABOVE CO_2, H_2O (VAPOR), O_2, AND AIR – N_2 AT 100°F

Q_v = INT. ENERGY OF COMBUSTION, AT 100°F., OF UNBURNED FUEL IN THE EQUILIBRIUM MIXTURE AT T. WHEN $T \leq 2880°R.$, $Q_v = 2.0, 2.5, 4.0$, AT $S = 0.4, 0.6, 0.8$, RESPECTIVELY

$E_s = E - Q_v$

H = ENTHALPY, $E + J(PV)$

$H_s = E_s + J(PV)$ $[\therefore H = H_s + Q_v]$

P = PRESSURE, LBS./SQ. IN. (DASHED LINES)

V = VOLUME, CU. FT. (SOLID LINES)

S = ENTROPY, ABOVE CO_2, H_2O (VAPOR), O_2, AND AIR – N_2, EACH AT ONE ATMOSPHERE, 100°F

T = TEMPERATURE, °R = (°F + 460)

FUEL = $(CH_2)_x$

APPROXIMATE SCALE FOR HORIZONTAL INTERPOLATION BETWEEN PRESSURE OR VOLUME LINES

H - INTERPOLATION SCALE

INTERNAL ENERGY, 100'S B.T.U.

$\frac{H_s}{100}$ $\frac{E}{100}$

$\frac{E_s}{100}$

DASHED LINES, PRESSURE

SOLID LINES, VOLUME

DASHED LINES, PRESSURE

SOLID LINES, VOLUME

DASHED LINES, PRESSURE

IN THIS RANGE,	
S	Q_v
0.3	1.5
0.4	2.0
0.5	2.0
0.6	2.5
0.7	3.0
0.8	4.0

F = 0.8
PRODUCTS OF COMBUSTION OF 1 LB. OF AIR WITH 80% OF THEORETICAL FUEL, IN CHEMICAL EQUILIBRIUM 0.05420 LBS. FUEL/LB. AIR

ALL VALUES OF EXTENSIVE VARIABLES TAKEN FROM THIS CHART ARE ON A MASS BASIS AND THEREFORE ARE REFERRED TO IN THE TEXT BY LOWER-CASE SYMBOLS.

ENTROPY, B.T.U. PER DEGREE R.

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M. I. T., CAMBRIDGE, MASS.

E = INTERNAL ENERGY, B.T.U., ABOVE CO_2, H_2O (VAPOR), O_2, AND AIR – N_2 AT 100°F

Q_v = INT. ENERGY OF COMBUSTION, AT 100°F, OF UNBURNED FUEL IN THE EQUILIBRIUM MIXTURE AT T. WHEN T ≤ 2880°R., Q_v = 1, 2, 5, AT S = 0.4, 0.6, 0.8, RESPECTIVELY

$E_s = E - Q_v$

H = ENTHALPY, $E + J(PV)$

$H_s = E_s + J(PV)$ [$\therefore H = H_s + Q_v$]

P = PRESSURE, LBS./SQ. IN. (DASHED LINES)

V = VOLUME, CU. FT. (SOLID LINES)

S = ENTROPY, ABOVE CO_2, H_2O (VAPOR) O_2, AND AIR – N_2, EACH AT ONE ATMOSPHERE, 100°F.

T = TEMPERATURE, °R = (°F + 460)

FUEL = $(CH_2)_x$

APPROXIMATE SCALE FOR HORIZONTAL INTERPOLATION BETWEEN PRESSURE OR VOLUME LINES

H - INTERPOLATION SCALE

INTERNAL ENERGY, 100'S B.T.U.

$\frac{H_s}{100}$ $\frac{E}{100}$

$\frac{E_s}{100}$

DASHED LINES, PRESSURE

SOLID LINES, VOLUME

DASHED LINES, PRESSURE

SOLID LINES, VOLUME

DASHED LINES, PRESSURE

IN THIS RANGE,	
S	Q_v
0.3	0.5
0.4	1.0
0.5	1.5
0.6	2.0
0.7	3.0
0.8	5.0

ALL VALUES OF EXTENSIVE VARIABLES TAKEN FROM THIS CHART ARE ON A MASS BASIS AND THEREFORE ARE REFERRED TO IN THE TEXT BY LOWER-CASE SYMBOLS.

F = 1.0
PRODUCTS OF COMBUSTION OF 1 LB. OF AIR WITH THE THEORETICAL FUEL, IN CHEMICAL EQUILIBRIUM
0.06775 LBS. FUEL/LB. AIR

ENTROPY, B.T.U. PER DEGREE R.

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M.I.T., CAMBRIDGE, MASS.

E = INTERNAL ENERGY, B.T.U., ABOVE CO_2, H_2O (VAPOR), O_2, AND AIR – N_2 AT 100°F.

Q_v = INT. ENERGY OF COMBUSTION, AT 100°F., OF UNBURNED FUEL IN THE EQUILIBRIUM MIXTURE AT T. WHEN T ≤ 2880°R., Q_v = 169.

$E_s = E - Q_v$

H = ENTHALPY, $E + J(PV)$

$H_s = E_s + J(PV)$ [$\therefore H = H_s + Q_v$]

P = PRESSURE, LBS./SQ. IN. (DASHED LINES)

V = VOLUME, CU. FT. (SOLID LINES)

S = ENTROPY, ABOVE CO_2, H_2O (VAPOR) O_2, AND AIR – N_2, EACH AT ONE ATMOSPHERE, 100°F.

T = TEMPERATURE, °R = (°F + 460)

FUEL = $(CH_2)_x$

APPROXIMATE SCALE FOR HORIZONTAL INTERPOLATION BETWEEN PRESSURE OR VOLUME LINES

H – INTERPOLATION SCALE

$\frac{H_s}{100}$

$\frac{E_s}{100}$

INTERNAL ENERGY, 100'S B.T.U.

IN THIS RANGE $E - E_s = Q_v = 169$

DASHED LINES, PRESSURE

SOLID LINES, VOLUME

DASHED LINES, PRESSURE

SOLID LINES, VOLUME

F = 1.1

PRODUCTS OF COMBUSTION OF 1 LB. OF AIR WITH 110% OF THEORETICAL FUEL, IN CHEMICAL EQUILIBRIUM 0.07453 LBS. FUEL/LB. AIR

ALL VALUES OF EXTENSIVE VARIABLES TAKEN FROM THIS CHART ARE ON A MASS BASIS AND THEREFORE ARE REFERRED TO IN THE TEXT BY LOWER-CASE SYMBOLS.

ENTROPY, B.T.U. PER DEGREE R.

$\frac{E}{100}$

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M.I.T., CAMBRIDGE, MASS.

E = INTERNAL ENERGY, B. T. U., ABOVE CO_2, H_2O (VAPOR), O_2, AND AIR - N_2 AT 100°F

Q_v = INT. ENERGY OF COMBUSTION, AT 100°F., OF UNBURNED FUEL IN THE EQUILIBRIUM MIXTURE AT T. WHEN T ≤ 2880°R., Q_v = 336.

$E_s = E - Q_v$

H = ENTHALPY, $E + J(PV)$

$H_s = E_s + J(PV)$ [$\therefore H = H_s + Q_v$]

P = PRESSURE, LBS./SQ. IN. (DASHED LINES)

V = VOLUME, CU. FT. (SOLID LINES)

S = ENTROPY, ABOVE CO_2, H_2O (VAPOR), O_2, AND AIR - N_2, EACH AT ONE ATMOSPHERE, 100°F

T = TEMPERATURE, °R = (°F + 460)

FUEL = $(CH_2)_x$

APPROXIMATE SCALE FOR HORIZONTAL INTERPOLATION BETWEEN PRESSURE OR VOLUME LINES

H - INTERPOLATION SCALE

DASHED LINES, PRESSURE

SOLID LINES, VOLUME

DASHED LINES, PRESSURE

SOLID LINES, VOLUME

DASHED LINES, PRESSURE

ALL VALUES OF EXTENSIVE VARIABLES TAKEN FROM THIS CHART ARE ON A MASS BASIS AND THEREFORE ARE REFERRED TO IN THE TEXT BY LOWER-CASE SYMBOLS.

F = 1.2
PRODUCTS OF COMBUSTION OF 1 LB. OF AIR WITH 120% OF THEORETICAL FUEL, IN CHEMICAL EQUILIBRIUM
0.08130 LBS. FUEL/LB. AIR

$\frac{H_s}{100}$ $\frac{E}{100}$

ENTROPY, B. T. U. PER DEGREE R.

$\frac{E}{100}$

INTERNAL ENERGY, 100'S B. T. U.

IN THIS RANGE $E - E_s = Q_v = 336$

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M. I. T., CAMBRIDGE, MASS.

E = INTERNAL ENERGY, B. T. U., ABOVE CO_2, H_2O (VAPOR), O_2, AND AIR – N_2 AT 100°F

Q_v = INT. ENERGY OF COMBUSTION, AT 100°F., OF UNBURNED FUEL IN THE EQUILIBRIUM MIXTURE AT T. WHEN $T \leq 2880°R.$, $Q_v = 336$.

$E_s = E - Q_v$

H = ENTHALPY, $E + J(PV)$

$H_s = E_s + J(PV)$ [$\therefore H = H_s + Q_v$]

P = PRESSURE, LBS./SQ. IN. (DASHED LINES)

V = VOLUME, CU. FT. (SOLID LINES)

S = ENTROPY, ABOVE CO_2, H_2O (VAPOR), O_2, AND AIR – N_2, EACH AT ONE ATMOSPHERE, 100°F

T = TEMPERATURE, °R = (°F + 460)

FUEL = $(CH_2)_x$

APPROXIMATE SCALE FOR HORIZONTAL INTERPOLATION BETWEEN PRESSURE OR VOLUME LINES

H - INTERPOLATION SCALE

DASHED LINES, PRESSURE

SOLID LINES, VOLUME

DASHED LINES, PRESSURE

SOLID LINES, VOLUME

DASHED LINES, PRESSURE

ALL VALUES OF EXTENSIVE VARIABLES TAKEN FROM THIS CHART ARE ON A MASS BASIS AND THEREFORE ARE REFERRED TO IN THE TEXT BY LOWER-CASE SYMBOLS.

F = 1.2
PRODUCTS OF COMBUSTION OF 1 LB. OF AIR WITH 120% OF THEORETICAL FUEL, IN CHEMICAL EQUILIBRIUM
0.08130 LBS. FUEL/LB. AIR

$\frac{H_s}{100}$ $\frac{E}{100}$

ENTROPY, B. T. U. PER DEGREE R.

INTERNAL ENERGY, 100'S B. T. U.

IN THIS RANGE $E - E_s = Q_v = 336$

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M. I. T., CAMBRIDGE, MASS.